MODERN
REFRIGERATING MACHINERY

ITS CONSTRUCTION, METHODS OF WORKING AND INDUSTRIAL APPLICATIONS

A GUIDE FOR ENGINEERS AND OWNERS OF REFRIGERATING PLANTS

BY

PROFESSOR HANS LORENZ, PH.D.

Principal of the Institute for Technical Physics, Göttingen University

AUTHORIZED TRANSLATION FROM THE THIRD GERMAN EDITION

BY

THOMAS H. POPE, B.SC.

Lecturer in the British School of Malting and Brewing, Birmingham University

WITH CHAPTERS ON AMERICAN PRACTICE IN REFRIGERATION, INSULATION, AUDITORIUM AND OTHER COOLING

BY

H. M. HAVEN, S.B.

Member of the American Society of Mechanical Engineers and of the American Society of Refrigerating Engineers

AND

F. W. DEAN, S.B.

Member of the American Society of Mechanical Engineers

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AUTHOR'S PREFACE TO THE THIRD EDITION.

THE preparation of this work, which is intended especially to meet practical requirements and which appeared first at the end of 1896, and as a second edition[1] in the beginning of 1899, has been urged upon me by several colleagues. I have attempted to treat of the province of the artificial production of cold and of its applications as they exist at the present time, but I have limited myself to European conditions which have come within my own experience.

The aim of the book forbids a mathematical treatment of the subject; those numbers arrived at theoretically and necessary to the understanding of the processes described, together with numerous physical data, hence only find a place in Tables. I have sought to explain the application of these results to the elementary, approximate solution of the most important practical problems by the insertion of numerical examples.

For a detailed and more especially theoretical treatment, with which the designer cannot dispense, I must refer the reader to text-books on Thermodynamics and to my papers: "*Beiträge zur Beurteilung von Kühlmaschinen,*" Zeitschrift des Vereins Deutscher Ingenieure, 1894; "*Vergleichende Theorie und Berechnung der Kompressions-Kühlmaschinen,*" Zeitschrift für die gesamte Kälteindustrie, 1897, and "*Die Wirkungsweise und Berechnung der Ammoniak-Absorptionsmaschinen,*" ibid., 1899.

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[1] Besides these a French translation has been prepared by P. Petit and J. Jaquet, and was published in 1897 by Messrs. Gauthier-Villars et Fils of Paris under the title "Machines frigorifiques."

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Great value must be attached to a systematic arrangement of the matter, and I hope that I have succeeded in compressing the comprehensive material at my disposal into a relatively narrow space without influencing its lucidity.

In view of the favourable reception accorded to the book in practical circles, I have in this *third edition* held fast to these principles. Besides numerous small additions, which are, for the most part, the immediate results either of experiment or of stimulating intercourse with practical engineers, this edition contains a section (Chapter X) on the determination of the cooling effect, which I hope will be welcome to many readers.

On the other hand, the list of references to the literature, which experience has shown me is made little use of, has been entirely omitted, owing to the demands of space. Finally, I wish to express my thanks to my assistant, Mr. G. Cattaneo, for his help in reading the proofs.

H. LORENZ.

GÖTTINGEN, June 1901.

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MODERN REFRIGERATING MACHINERY.

CHAPTER I.

THE PRINCIPLES OF HEAT.

1. **Temperature.**—The problem of the *production of cold* has, in general, two parts: first, the *lowering of the temperature* of a solid, liquid, or gaseous body below that of its surroundings, and, further, the *maintenance of this low temperature* against the action of external and internal influences which strive to restore the original state of things. In the *heating* of a body similar relations hold, except that here we have to do with the attainment and preservation of *temperatures higher* than that of the surrounding objects. In both cases we have to combat the striving of nature after *uniformity of temperature.*

In order to understand the problems met with in the production of cold, it will be necessary to learn something of the processes with which such a levelling of temperature is connected, and, first of all, to obtain a clear idea of what is meant by *temperature.* This has arisen from our sense of feeling or touch, which makes us sensitive to the difference between the actions on our skin of bodies with high and of those with low temperatures. If, now, we bring two such bodies into contact with one another, we find that the one appearing to us to be the warmer contracts, whilst the other expands, these two actions continuing until our sense of touch is no longer able to discern any difference between them. If, however, instead of our skin, we make use of another body, e.g., a mercury column enclosed in a narrow glass tube of uniform bore, we find that in contact with the warmer body this column increases, and with the colder one decreases, in length; after the two bodies have become uniform in temperature, the mercury column assumes a length intermediate to those obtained in the other two cases. If, then, we use the length of this mercury thread as a measure of the temperature and take care that, as with the two observed bodies, its temperature has been levelled by contact, we arrive at the important experimental law: *Two bodies have the same temperature, if a third—the so-called thermometer—when brought separately into contact with them under constant conditions, assumes one and the same volume.* In order to obtain a starting-point for our temperature measurement, we place the thermometer in contact with melting ice and mark the position of the end of the corresponding mercury column. This point will always be reached by the mercury thread if we repeat this procedure, for instance, with different ice or at another place, supposing only that the atmospheric pressure (barometric height) is always the same. We may, therefore, use this position of the end of the thread as a *standard point* of our scale. Another point may be obtained by immersing the thermometer in boiling water, and this will always be subsequently obtained, provided that the atmospheric pressure on the boiling water remains constant. The pressure chosen is that of a mercury column 29·92 inches high, the *boiling-point* being then marked on the thermometer. We then divide the space between the freezing- and boiling-points into a number of equal parts or so-called *degrees* (into 180, according to the suggestion of *Fahrenheit*), and continue this division both below the freezing-point and above the boiling-point; in this way we arrive at a *scale for temperature independent* of the manifold variations of our sense of feeling and also of the dimensions *of the thermometer.* As may be seen from the method by which it was arrived at, this scale is perfectly arbitrary, but it attains a more general physical signification if we investigate with it the behaviour of certain gases, for example, ordinary air. Under constant pressure these gases expand, for every degree their temperature is raised, by $\frac{1}{492}$ of the volume they possess at the freezing-point (32° F.), and contract to the same extent for a corresponding lowering of temperature (*Gay-Lussac's* Law). The value $\frac{1}{492}=0 \cdot 00203$ we term their *coefficient of expansion*, which for these gases is to be considered as invariable and quite independent of our scale of temperature. If we neglect small deviations, which do not interest us at first, and assume that the above law holds also for the very highest and lowest temperatures, it follows that at 492° F. below the zero-point (– 460° F.) the volume of a gas will diminish to nothing. In spite of the fact that such a condensation of material bodies to a point is in the highest degree improbable and is excluded by the continually increasing influence of the above-mentioned deviations from the gas law, yet there is sufficient reason for imagining the lowest conceivable temperature to be in the neighbourhood of – 460° F., and we shall therefore take this temperature to be the *absolute zero* or the *starting-point of the absolute scale of temperature.* Fahrenheit temperatures of – 460°, – 100°, 0°, + 32°, and + 100° F. will then correspond with absolute temperatures of 0°, 360°, 460°, 492°, and 560° respectively.

2. **Heat.**—The changes taking place between two bodies, initially of different temperatures, when brought into contact, are not limited to their temperatures becoming identical. When this change takes place between two equally large quantities of water at different temperatures, the final temperature will be the arithmetic mean of the two original ones. If, however, the quantities of water are unequal, e.g., 1 lb. at 212° F. and 3 lbs. at 68°, the end temperature of the total mass of 4 lbs. will be 104°, since 1 lb.×212°+3 lbs.×68°= 4 lbs.×104°, so that there has taken place between the two masses of water a transference of

$$1 \text{ lb.}(212-104)^\circ = 3 \text{ lbs.}(104-68)^\circ = 108 \text{ pound-degrees.}$$

It is seen from this that the final temperature on mixing does not depend on the initial temperatures alone, but also on the quantities mixed. That which is lost by one of the bodies and gained by the other we term *heat*, which, as in the above example, is measured by the rise in temperature of a certain quantity of water. By a *Unit of Heat* or *British Thermal Unit* (B.T.U.) we understand the quantity of heat[1] necessary to heat 1 lb. of water by 1° F.

If, now, we bring another body, such as a metal, into contact with water, the change in temperature of the latter will, in general, be different from that caused by the addition of water of the same weight and temperature as the body. The best method of procedure is to indicate the quantity of water which, with the same initial temperature, would give the same result as the body, and to term this the *water-equivalent* of the body. But since this water-equivalent, according to our definition of the unit of heat, indicates how many such units are necessary to heat a certain weight of the body by 1° F., we obtain, on dividing by the weight of the body, the quantity of heat required to heat 1 lb. of the body 1° F., or the so-called *specific heat.* Knowing the magnitude of the water-equivalent for any body subjected to heating or cooling, it is only necessary to multiply it by the rise or fall in temperature in order to arrive at the heat added to or subtracted from it.

For example, imagine a vessel of sheet iron weighing 1000 lbs. and containing 7000 lbs. of a salt solution of specific heat 0·83, the iron having a specific heat of 0·11. At the beginning of the measurement the vessel and contents had a temperature of 32°, and at the end 23°; the amount of the heat lost is required.

First, the *water-equivalent* of the vessel and contents is given by $1000 \times 0.11 + 7000 \times 0.83 = 5920$, so that the heat lost will be $5920 \times 9 = 53,280$ B.T.U.

The experiment which we imagined carried out in order to arrive at the conception of specific heat may be directly employed for the determination of this magnitude for different bodies. In doing this, the temperatures between which cooling or heating takes place should be accurately noted, because it is found experimentally that the specific heat of a body is not constant under all conditions. For technical purposes, however, this variation may be neglected in the case of solids or liquids, if the temperature-interval is not too great and is situate sufficiently far from the solidifying- or boiling-point.

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[1] To be quite accurate the limits of temperature within which the measurement is made should be stated, but for technical purposes the definition here given is sufficient.

With gases and vapours, on the other hand, other influences come into play, and these we shall consider later.

All bodies show peculiar behaviour in their so-called *change of state*, i.e., in their passage from the solid to the liquid condition (melting) or from the liquid to the vaporous state (evaporation) or inversely (solidification, condensation). If, for instance, a certain quantity of ice is heated in a vessel, its temperature rises to 32° F., after which it begins to melt without further rise of temperature, in spite of the addition of a considerable quantity of heat (about 144 B.T.U. per lb.). Only after all the ice has been turned to water does the temperature rise again, and then it proceeds to the boiling-point, which under atmospheric pressure is 212° F. Then, without further rise in temperature, the water begins to evaporate, the absorption of heat (about 1080 B.T.U. per lb.) being even more considerable than in the process of melting. These quantities of heat, used up during melting and evaporation, are set free on solidification and condensation and are, like the corresponding temperatures, dependent on the external pressure; they are termed the *latent heats of liquefaction and vaporisation*. In the production of cold these processes are of the greatest importance, since they are attended by the absorption or evolution of extremely large quantities of heat.

3. The Equivalence of Heat and Mechanical Work.—We have up to now regarded heat as a phenomenon of nature without any connection with other forms of energy. That such a connection exists has been, however, long known from the fact that when mechanical work is destroyed, e.g., by the rubbing together of solid bodies, heat is produced. That these two magnitudes bear to one another a *definite relation, which is quite independent of the nature of the conversion,* was first shown by accurate measurements in the year 1842 and led to the perception of the law of the *equivalence of heat and work* (Law of *Mayer* and *Joule*). The generation of 1 B.T.U. was found to correspond with the destruction of about 778 foot-pounds of work. By this means heat was at the same time recognised as a *form of energy* and it became possible to compare mechanical and thermal processes quantitatively. If we understand by *energy* the total latent heat stored up in a body together with that made manifest by its temperature, we may say that *the addition of a certain quantity of heat to a body corresponds with the sum of the increase of its energy and the equivalent of the work yielded by it during the absorption of heat.* For a given addition of heat there is a large number of conceivable *alterations of state* varying with the nature of the work performed. The study of these forms the subject of the *mechanical theory of heat* or *thermodynamics,* the *first law* of which is the equivalence of heat and work.

A simple application of this law may be made in the case of gases. If, for example, 1 lb. of air at 32° F. is enclosed in a cylinder of 1 sq. ft. sectional area by means of a movable, frictionless and air-tight disc (Fig. 1), the load on the latter and hence on the gas will be that of the atmospheric pressure, namely, 2117 lbs. per square foot. But the volume of 1 lb. of air at 32° F. is 12·37 cub. ft. If, now, the air in the cylinder is heated by 1° F., its volume (see above, § 1) will increase by $\frac{1}{492}$, so that the disc will be displaced to the extent of $\frac{12\cdot 37}{492}=0\cdot 0253$ ft., by which means $0\cdot 0253\times 2117=53\cdot 48$ foot-pounds of work, corresponding to a heat equivalent of 0·069 B.T.U., are done against the atmospheric pressure. The heat necessary

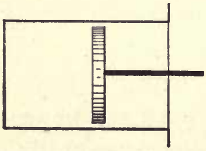

FIG. 1.

to raise the temperature 1° F., which is termed the *specific heat at constant pressure*, is 0·238 B.T.U., of which 0·069 B.T.U. is used up in doing external work. It may therefore be concluded that, at constant volume (with a fixed disc), only 0·238 − 0·069 = 0·169 B.T.U. would be required In fact, as other investigations show, this is the value of the *specific heat* of air *at constant volume.*

Further researches, due mainly to the French worker *Regnault*, have shown that these specific heats of gases possess nearly *constant values*, so that *the energy-content of 1 lb. of gas is, within wide limits, obtained simply by multiplying its absolute temperature by the specific heat at constant volume.*

The work performed in the expansion of the gas to the volume corresponding with the temperature is not taken into account in the energy-content, since it serves for overcoming external resistance (i.e., the atmospheric pressure) and is hence no longer contained in the gas. This experimental fact has a certain significance in the production of cold, when taken in connection with the law discovered by *Boyle* and by *Mariotte*, which states that the *pressure*[1] *acting on a gas multiplied by the volume it occupies has a constant value at one and the same temperature.* But, as we have already seen (§ 1), the volume increases proportionally with the absolute temperature, so that *the product of pressure and volume, and consequently also the energy-content of the gas, are proportional to this temperature.* A good idea of this behaviour is obtained from a graphic representation of the relation between pressure and volume at different temperatures, as shown in Fig. 2. The curves thus obtained, each corresponding with a definite temperature, are known as the *isotherms* of the gas in question. If a gas expands isothermally, i.e., so that its pressure and volume always follow such a curve, and if in so doing it yields a certain amount of external work, then, in order that the temperature may remain constant, the heat-equivalent corresponding with this work must be supplied to it.

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[1] By pressure is here meant always the absolute and not the excess pressure.

If this does not take place, a lowering of temperature will occur corresponding with the work done, and when this is in any degree appreciable the fall in temperature may be very considerable; so that, if the initial temperature of the gas is not much above the melting-point, that finally attained will be very much below the latter. Conversely, the compression of a gas must be attended with a rise of temperature if the heat-equivalent of the work of compression is not removed.

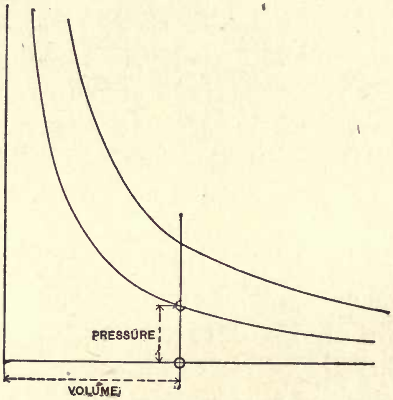

FIG. 2.

If, on the other hand, a gas confined under constant pressure is allowed to *flow out* into a space at a lower but also constant pressure, then, after the escaped portion has come to rest again, the work transferred to this portion from the enclosed gas is equal to the work of displacement it itself does, so that *no change in the energy-content and hence in the temperature can take place.* This consequence of the above-developed law was tested by *Thomson* and *Joule* in England and found to be correct, except for small deviations which have recently been made technically useful.

As follows from the above considerations referring to gases, the *state* of a body, especially its *energy*, is completely determined *by the pressure and its volume,* since these two magnitudes fix also the temperature (see the isotherm-diagram, Fig. 2). Further, it is also evident that a body in the condition A can be transferred to another state B in very different ways; e.g., it may be first compressed without the addition or removal of heat and then have heat added to or subtracted from it, or these two operations may be combined, etc. If this process is carried out in one way and the body then allowed to expand again under other conditions (see Fig. 3) until it returns to its original state, it is evident that the body has passed through a cycle of processes at the end of which its energy has the same value as at first. *The heat-equivalent of the work yielded by such a cycle must therefore be equal to the difference between the amounts of heat added and subtracted.* If the amount of heat subtracted is greater than that added, work must be done during the process.

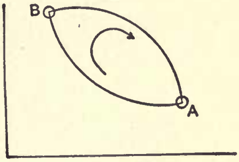

FIG. 3.

Processes of the first kind, which yield work, are made use of in motors, and those of the second kind, which require an expenditure of work, in cooling machinery. In both cases we have, according to the law just given, to deal with quantities of energy supplied and removed, and these must be compared, like the credit and debit of the bookkeeper, in order to obtain the balance, which is here known as the *balance of heat.*

4. The Convertibility of Heat.—While experiment has shown that it is possible, under all circumstances, to convert a certain quantity of work completely into heat, the reverse change cannot, in general, be carried out. In all processes practically possible, only a part of the heat is transformed into work, whilst the rest of the heat applied reappears as heat and at a lower temperature than before the work is done. As an example, we may recall the steam-engine to which heat at a high temperature is introduced by means of steam. After the work is done, the heat, diminished by the equivalent of the work, is transferred to the condenser, the temperature of which is considerably lower than that of the steam in the boiler. It is now undoubtedly desirable to ascertain what proportion of the heat taking part in such a passage from a high-temperature level to a lower one can, under the best conditions, be converted into work.

For this purpose we will make use of a mechanical analogy for the process devised by *Zeuner.* Since heat is equivalent to mechanical work, we may represent it by a weight multiplied by the height raised. The height must evidently be represented by the difference between the absolute temperatures, so that the

$$\text{Weight of Heat}=\frac{\text{Quantity of Heat}}{\text{Absolute Temperature}}.$$

If we imagine that this ideal weight falls from a temperature-level T_1 to a lower one T_2, corresponding to the earth, in the case of a falling body, then, if no part of the weight is lost on the way (as by the radiation of heat), the heat-weight will be the same after the fall as before (Fig. 4). We have consequently the law that, in the transference, under the most favourable circumstances, of a quantity of heat Q_1 from a high-temperature level T_1 to a lower one T_2 with the performance of L foot-pounds of work so that the quantity of heat $Q_2=Q_1-L/778$, *the two quantities of heat* Q_1 *and* Q_2 *bear the same relation to one another as the corresponding absolute temperatures.*[1] (Law of *Carnot* and *Clausius,* or the *Second Law of Thermodynamics.*) If the fall is accompanied by the performance of no useful work, i.e., if the weight simply drops, the whole of the energy reappears as heat when it stops, just as in the transference of heat from a body with a high temperature to one with a lower temperature.

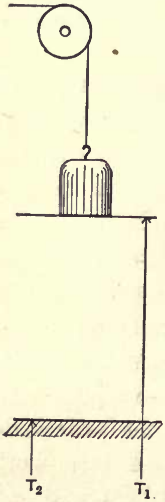

FIG. 4.

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[1] It may as well be expressly stated that the above representation is not a proof, but only an illustration of the second law of thermodynamics.

But for the *production of cold* the *reverse operation* is of the greatest importance. Just as it is impossible to move a weight from a low level to a higher one without the expenditure of work, so also the corresponding conveyance of heat from a cold body to a hotter one requires the application either of a quantity of work at least as great as that obtainable from the reverse change or of the heat-equivalent of this work. The heat given up to the hotter body is then greater by this equivalent than the amount taken from the colder body. Further, in these processes losses are possible, on the one hand owing to the velocities not being equalised when the lifting is finished, and on the other from additional loads (corresponding to radiation of heat) put on during the lifting process, so that the work obtained above always represents the *smallest amount, if the temperatures of the bodies giving up and absorbing the heat do not change.*

If, however, such changes of temperature do occur, the minimum work is obtained sufficiently accurately by means of the average temperatures of the two bodies.[1]

For the production of cold, intermediate bodies, so-called *cooling-media,* or *cooling-agents,* are employed, these being able to take up large quantities of heat at low temperatures and to give them up again at higher temperatures. If the temperatures of the cooling-agent do not appreciably change during the taking in and giving out of the heat, it is convenient, for making approximate calculations, to introduce these temperatures in place of those of the bodies actually taking up and giving out the heat.

For the sake of clearness we may give numerical examples. From a salt solution at 14° F., i.e., 474° absolute, 100,000 B.T.U. (Q_2) are to be removed hourly and are to be transferred to cooling water at +50°, then +68°, and finally +86°, i.e., at 510°, 528°, and 546° absolute. The heat to be given up hourly is calculated by means of the values of the ratio $Q_1:Q_2$—namely, 510:474, 528:474, and 546:474—to be 107,600, 111,400, and 115,200 B.T.U.; the necessary work to be done is then obtained by multiplying the equivalents 7600, 11,400, and 15,200 B.T.U. by 778, giving 5,910,000, 8,860,000, and 11,820,000 foot-pounds, or, since 1 horse-power represents 1,980,000 foot-pounds per hour, 2·99, 4·48, and 5·97 horse-power.

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[1] To such cycles, which, with variable upper and low temperatures, yield or require the greatest amount of work, I have given the name polytropic; these have been treated in detail in several theoretical papers (Zeitschrift des Vereins deutscher Ingenieure, 1894, and Zeitschrift f. Kälteindustrie, 1895).

The processes here considered are quite independent of the nature of the cooling-agent. This is, however, only true so long as the process by which the transference of heat is effected remains perfectly *reversible*, i.e., it can be interrupted at any time and allowed to proceed in the opposite direction. If, indeed, with some other body another result was arrived at, it would only be necessary to carry out the transference of heat from a higher to a lower level with a body which gives out more work than is required by another in the reverse process, in order to obtain perpetual motion. The impossibility of this leads, however, to the important law that the body used would exert no influence on the final result if we were in a position to overcome the production of cold completely by a reversible cycle of processes. Among the latter, *Carnot's cycle* occupies a special position, because by the help of saturated vapours as media we are able to approach very near to it. If, for example, we allow a liquid, formed at a normal temperature (between +50° and 86° F.) by condensation from a vapour, to give up its energy in a cylinder until its temperature has sunk to some required lower level, we can evaporate it at this temperature and the corresponding low pressure by applying the heat Q_2 (cold produced) and can then draw it through a compressor and bring it up again to the high pressure or the high temperature without the application of heat;[1] finally, by withdrawing another quantity of heat, Q_1, it may be converted once more into a liquid. In this way we obtain a diagram such as that given in Fig. 5, where the singly hatched surface represents the ideal amount of work required, while the doubly hatched one shows the work which can be again obtained from the cylinder by means of the liquid.

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[1] Such a change of state is termed either *adiabatic*, because while it is being executed no heat is supplied or withdrawn, or *isentropic*, owing to the constancy of the heat-weight (entropy).

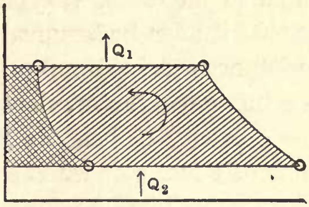

Fig. 5.

To this cycle correspond also the numerical examples given above. The most important deviation from the practical attainment of this is first the omission of the above-mentioned cylinder, due with some cooling-agents to the small yield and with others to certain difficulties of construction. The engines of the compression system actually used will therefore require a supply of energy greater by at least the work of this cylinder.

5. Conduction and Radiation of Heat.—In §§ 1 and 2 we have seen that, when two bodies of different temperatures come into contact, that having the higher temperature gives up heat to the other. Exactly the same process takes place between two parts of one and the same body which differ in temperature, and this continues until perfect uniformity of temperature has been attained. If, however, we maintain two places at constant but unequal temperatures and so prevent uniformity of temperature by supplying and withdrawing heat at the two parts, it is evident that a steady stream of heat will flow from the hotter to the cooler part of the body. Further, this stream will be maintained if we make a chain of bodies, that at one end being kept at a different temperature from that at the other. Such a process, which is known as *conduction of heat,* goes on everywhere where transference of heat takes place. When the difference of temperature is only a small one, as is usually the case in the production of cold, *the quantity of heat set in motion by conduction is proportional both to the sectional area of the body perpendicular to the direction of flow and also to the temperature-difference divided by the length of path.* If we experiment with several bodies, we find that, although the temperature-difference, cross-sectional area, and distance of flow are the same in all cases, varying quantities of heat are transferred, so that it must be assumed that the various bodies offer different resistances to heat-motion. Corresponding with this resistance is a numerical factor, the so-called *heat-conductivity*, which gives the quantity of heat in B.T.U. passing every hour through a cube of 1 foot side when one of the faces (of 1 sq. ft.) is exactly 1° F. hotter than the opposite one. According to the magnitude of the conductivity we distinguish *good* and *bad heat-conductors*, or briefly, *conductors* and *insulators;* the latter are used industrially in order to diminish as much as possible the passage of heat where this is undesirable.

Conduction of heat depends on the existence of difference of temperature and always takes place from the high to the low temperature level. Since a reverse flow of heat is impossible, the conduction of heat is termed an *irreversible process*. It is of importance here that the heat performs no work on its path, but reaches the lower level unchanged in quantity. If we wish to bring it to a high level again, i.e., to bring about a transference of heat from a colder to a warmer body, we must supply energy. This fact is comprehended in the law given by *Clausius*, which states that *transference of heat from a colder to a warmer body cannot take place without compensation.*

In practice such irreversible processes are never completely avoidable, so that the attainment of a perfectly reversible cycle is impossible. And if we use the latter as a standard of comparison, it is because, on the one hand, the fall of temperature required to bring about conduction of heat may be diminished as much as we like by increasing the section of flow, whilst, on the other, where conduction is inconvenient it can be diminished to any extent by insulation.

Further, between bodies which have different temperatures but are not in contact with one another a transference of heat is found to take place, and to this we give the name *heat-radiation.* This process could only go on quite alone in an absolute vacuum and proceeds so that the heat is changed at the surface of the hot body into radiant energy (which differs from light and electric waves solely in the wave-length), which is converted back into heat at the surface of the cold body. But since the space intervening between such bodies is always filled with a gas, e.g., air, this absorbs a part of the rays passing through it and changes their energy into heat, so that the whole of the radiated heat never reaches its goal. This absorption of heat-rays is of little importance in technical applications, where we are only interested in those amounts of heat which a hot body actually gives up to, or a cold body receives from, its surroundings. The quantitative treatment of these processes presents, too, considerable difficulty, so that *Kirchoff* has characterised the introduction of a so-called *coefficient of external conductivity*—by which both the radiation and also the conduction of heat through the air are taken account of—as arbitrary and hence misleading.

Radiation of heat into and out of bodies can only be guarded against by the use of surfaces which admit of only an extremely small absorption of heat, such as a metallic mirror or a white coating.

CHAPTER II.

METHODS OF COLD PRODUCTION AND ENERGY REQUIRED.

6. Methods of Cold Production.—The production of cold as at present practised is based almost exclusively on the rapid absorption of heat owing to the *evaporation* of more or less volatile liquids (condensed gases) *at low temperatures*. The method—at one time common, but now rarely used—of cold production by the cooling of air as it expands and performs external work in so-called *cold-air machines* has been found to be very uneconomical, because, owing to the low heat-capacity of air (only 0·2377 B.T U. per 1 lb., or roughly 0·034 B.T.U. per cubic foot for 1° F.), the lower limit of temperature must be made very low—which increases the amount of work necessary—and large quantities of air must be treated, so that immense machines must be employed and hence enormous frictional losses incurred.[1] Further, the steadiness of running was influenced by the unavoidable formation of snow from the moisture of the air, and the latter, when used direct from the machine for cooling purposes, often contained lubricating material in suspension. By the choice of a suitable circulating liquid—the so-called *cooling-agent*—these disadvantages may be removed, either completely or nearly so in the *cold-vapour machines* based on the principle of evaporation, so that these have now entirely replaced the cold-air machines and there is hence no need for us to describe in detail the construction of the latter, which was often very ingenious.[1]

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[1] These machines consisted always of a *compressor* for compressing the air, a *cooling apparatus*, in which the air could give up its heat of compression to cold water and have its temperature lowered nearly to that of the water, and finally of an *expansion cylinder*, in which the air did work and assumed a temperature low enough to enable it to take up considerable quantities of heat. The compressor and the expansion cylinder were usually connected by means of a crank-shaft, unless, indeed, their pistons were worked by a common rod. Since the work furnished by the expansion cylinder, which was provided, like a steam-cylinder, with a reversing arrangement, was not alone sufficient to drive the compressor, the difference had to be made up by a motor, generally attached to the same crank-shaft.

Details of these older machines are given in a paper by A. C. Kirk, "On the Mechanical Production of Cold" (Proceedings of the Institution of Civil Engineers, 1874). For later investigations see *Schröter*, "Untersuchungen an Kältemaschinen verschiedener Systeme," I. Bericht. Munich, 1887.

The cold-vapour machines, with which alone we have to concern ourselves in what follows, are divided into three groups:

I. *Pure compression machines*, in which perfectly anhydrous *ammonia*, *carbon dioxide*, or *sulphur dioxide*, after evaporation in a tubular apparatus—the so-called *evaporator* or *refrigerator*—is sucked off by a *compressor*, compressed and liquefied by water-cooling in a second tubular apparatus (*condenser*), finally passing again into the evaporator through a *regulating-valve* inserted to throttle the difference of pressure.

II. *Pure absorption machines*, in which the cooling agent consists exclusively of *ammonia*, which is evaporated and then absorbed by water; this solution is brought by a small pump to a high pressure, at which the gas is driven out of solution again by immediate application of heat (ordinary steam-heating) and finally, after freeing as far as possible from suspended ammonia solution, liquefied by cooling-water and returned through an overflow-valve to the evaporator to again take up heat.

III. *Combined machines*, in which, as in II, the cooling agent consists of *ammonia*, this being sucked from the evaporator by means of a compressor (as in group I) and compressed to the corresponding saturation pressure, without any changes being made in the absorption machine.

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[1] These old-fashioned machines are occasionally described in detail in some of the latest handbooks, without the authors daring to pass the death-sentence on them for their notoriously small value.

Another combination of this kind is met with in *water-vapour cooling-machines,* which are also known as *vacuum machines,* because in them water-vapour is formed from a salt solution at a low temperature and under the corresponding vacuum, and is then conveyed partly into the air by means of the air-pumps used for maintaining the vacuum (0·04 to 0·08 inch of mercury), but is mainly taken up by sulphuric acid in a special compartment (the absorber). In order to use the acid again it must be subjected to concentration, which, in the older machines of this type, now quite done away with, was carried out by steam-heating; in the more recent types, however, the acid is brought into direct contact with the hot waste gases from a coke fire, after which the heated acid is cooled with water and introduced again into the absorber. The water thus removed from the salt solution and passed out of the machine must, of course, be replaced, so as to maintain constancy of conditions.

Of these different systems, the pure absorption machine —developed mainly by *Carré*—was the first to find wide-spread industrial application, but after the compression system had been brought to greater perfection by *Pictet* and *Linde* it became very largely replaced, so that with us absorption machines are only rarely found working at the present day, although in hotter climates they are still retained on grounds with which we shall become acquainted later. The combination of the absorption machine with a compressor, described under III and suggested by *A. Osenbrück,* has not emerged from the experimental stage, while the vacuum machine in the form proposed by *Windhausen* has failed owing to the difficulties both of the chemical action of the sulphuric acid and also of the intermittency of the action. Indeed, for the moment, the *pure compression machines* have almost exclusive possession of the field, so that in what follows we shall have to deal mainly with this class of machine and with the practical requirements of safety of running and of regularity in the maintenance of temperature.

In Fig. 6 is shown diagramatically the relation between the principal parts of the compression cold-vapour machine. The *compressor* P serving for compressing the vaporous cooling agent is, in reality, an air-pump furnished with an automatic valve (more rarely with a slide). The *suction-valves* are shown at S_1, S_2, and the *pressure-valves* at D_1, D_2.

The ordinary *pressure-main* CC leads from the compressor to the *condenser* K, where it connects with a spiral surrounded by cooling-water entering at A and escaping at B. A stirrer (not shown in the figure) in the interior of the condenser produces continual motion of the water, which is provided by a pump. The passage of the cooling-agent—liquefied in the condenser—into the spiral of the *evaporator* V takes place through the *regulating-valve* R. The evaporator V is exactly similar in structure to the condenser, contains also a stirring arrangement, and is, in general, traversed from E to F by a salt solution not readily frozen. After cooling in the evaporator, this

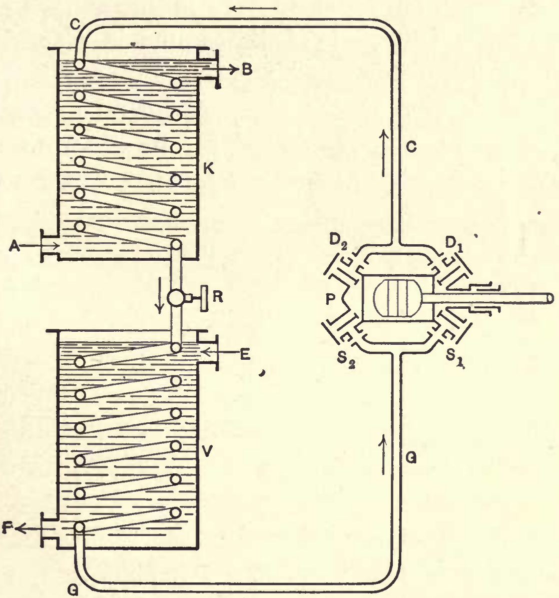

FIG. 6.

salt solution is transferred by special pumps to the place where it is to be used, e.g., the brewery cellar, and returns warm to the evaporator, while the cooling-agent, which enters the latter in the form of vapour, passes through the *suction-pipe GG* to the compressor *P*, where it begins a new circuit.

The construction of the *absorption machine* is considerably more complicated, because here not only the cooling of the condenser but also a quite separate removal of the heat of absorption must be provided for, in order that the water serving for the transport of the ammonia from the evaporator to the condenser shall not be influenced as regards its power of absorption in the absorber by an excessive rise in temperature. In the simplest form of absorption machine (Fig. 7) the *condenser C* receives the hot vapours driven off from the solution (of ammonia) in the *heater K*, cools them down and liquefies them as in the compression machine. Here also the passage into the evaporator *V* is effected by a regulating-

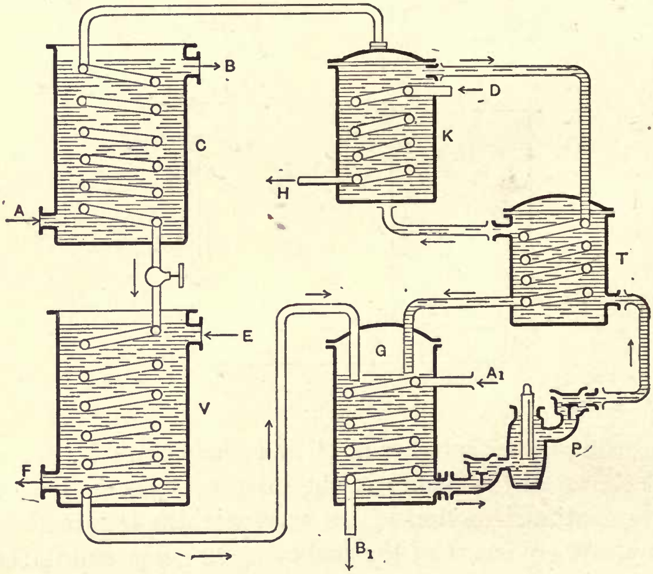

Fig. 7.

valve *R*. The ammonia is vaporised again by the absorption of heat at a low temperature and is then led into a vessel *G*, the so-called *absorber*, in which it unites with the weak solution flowing in from the heater, the heat of absorption being removed by cold water entering a spiral at A' and escaping at B'. The strong solution so formed is forced by a pump *P* into the heater *K*, the ammonia being here expelled from it by a steam-coil *DH* and again led into the condenser, the cycle of operations then beginning anew. This process would require the hot, weak solution coming from the heater to be cooled somewhat to the lower absorption temperature by means of cooling-water; on the other hand, the cold, strong solution taken from the absorber must also be brought in the heater to that temperature at which evolution of ammonia commences. Both processes can hence be combined in a so-called *interchanger*, which effects the heat exchange between the liquids flowing in opposite directions.

From this short description it will be seen that, in absorption machines, there are performed two cycles of operations which partially overlap. Besides the cycle of the ammonia there is that of the solution- or transport-medium through absorber, pump, interchanger, heater, interchanger, absorber, by which heat from outside is introduced into the heater but removed in the absorber. In Fig. 7 the pipes carrying the liquid corresponding with the second cycle are hatched horizontally throughout. Since the generation of cold, i.e., the transport of heat from the evaporator to the condenser, requires an expenditure of energy considerably greater than the work yielded by the pump (see § 10), the heat supplied to the heater must exceed that removed in the absorber by about the deficient amount of energy, for in the interchanger there only takes place an exchange within the apparatus. The heater corresponds, then, with the boiler of a steam-engine plant used to drive a compression cooling-machine, and the absorber with the condenser, while the interchanger plays to some extent the part of a feed-water heater.

A further improvement of this process has recently been suggested, consisting of the introduction of a second *interchanger* for the *vapours* evolved from the heater and from the liquid interchanger. These vapours are then passed through a system of pipes in a direction opposite to that taken by a part or the whole of the cold solution from the absorber—before it enters the first interchanger—so that they reach the condenser with a somewhat lower temperature than when they go direct from the heater. In this way not only is the cooling-water required for the subsequent cooling of these vapours economised, but also the further useless heating of the vapour developed in the interchanger by passing through the heater avoided.

On account of their low temperature, the evaporator- and also the suction-tubing of both systems must be protected against the intrusion of undesirable heat from outside, by means of *insulation*, i.e., a covering of badly-conducting substances.

7. Chemical Properties of the Most Important Refrigerants.—The chemical properties of the substances used in cooling machinery for carrying away the heat require especial attention, on account of their influence on the life of the whole installation. There comes also into play the possibility of the working becoming unsafe, owing to chemical actions. We have already seen that the association of these two inconveniences in the vacuum or water-vapour cooling-machine, in which the extremely chemically active sulphuric acid (H_2SO_4) is used as absorbent, would be a fatal objection. A short examination of the chemical properties of the principal substances employed at present as cooling-agents is therefore advisable. These are, as we have already seen, *ammonia* (NH_3), *carbon dioxide* (CO_2), *sulphur dioxide* (SO_2), and *water* (H_2O), the first being used alone in compression machines and together with water in absorption machines, while in some cases Pictet's suggestion to use carbon dioxide mixed with sulphur dioxide was formerly carried out. At the present time these two substances are employed separately in special machines which we shall consider later.

Ammonia, carbon dioxide, and sulphur dioxide, within wide limits of temperature and at the atmospheric pressure, are gaseous compounds and, in the pure state, are practically without action on the carbon and iron alloys (cast iron, steel, wrought iron) so largely employed for machinery. This is surprising with such a powerful base as ammonia and with the relatively strong sulphur dioxide, but carbon dioxide is noted for its chemical inactivity. Metals—especially copper and its alloys (bronze, brass)—are only attacked by ammonia in presence of oxygen (air) or water, the introduction of which into the machines must as far as possible be avoided. In quite small proportions, such as occur in commercial liquid ammonia, such admixtures are not dangerous. It is, however, best to dispense entirely with the use of copper in ammonia machines, since loose, non-compact places in the metal are scarcely to be avoided and these afford opportunity for the simultaneous action of air and water-vapour. The danger of explosion of ammonia,[1] which is so greatly feared, especially in America, has no real existence, since the decomposition of ammonia and the consequent liberation of the explosive hydrogen only takes place at a red heat. Such explosions are therefore only possible in case of fire, where the larger parts of the machine become heated to redness, and are of no importance in normal working.

Nevertheless it is important to be able readily to detect any impurities in the ammonia, both before its introduction into the machine and during the working. For this purpose the most suitable apparatus is one introduced by the Linde Ice Machine Co. and consisting of a cylindrical glass vessel of about 1 inch diameter, open at the top and provided with a graduated tube 0·4 in. wide and 0·2 cub. in. in capacity, fused into the bottom. The whole vessel contains about 2 cub. in.; it is filled with the liquid ammonia, which is then left to evaporate. The evaporation may be accelerated or more thoroughly carried out by slight heating (by placing the vessel on a steam-engine cylinder). The residue, which the researches of *Bunte* and *Eitner*[1] and of *Lange*[2] have shown to consist of water, alcohol-like substances, and organic bodies of high boiling-point, admits of an estimate being formed concerning the degree of impurity.

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[1] Siebel, "Disturbances in Cooling Machinery," Ice and Refrigeration, 1894, and Zeitschrift für Kälteindustrie, 1894.

It may be here remarked that the organic substances probably have their origin either in the lubricating-oil used in the compression pump in the manufacture of the ammonia, or in the compressor-oil of the cooling-machine itself, while the water is either introduced during the working owing to the oil-pump not being proof against drops, or else represents residues left in the coils after the evaporation. Both evils can, of course, be easily avoided by careful working.

The impurities of *carbon dioxide*, detectable especially by the smell, have been studied by *Grünhut.*[3] In artificially prepared carbon dioxide, subsequently compressed and liquefied, he found considerable quantities of ferric hydroxide, chloride, and sulphate. These compounds have doubtless arisen in the preparation of carbon dioxide in presence of water and air or of acids, such as hydrochloric and sulphuric. If such acids are present, even in traces, in the carbon dioxide introduced into the machine, they may exercise a powerful disturbing action. It is therefore advisable, when possible, to use for cooling machinery only natural carbon dioxide, which is found to be free from such admixtures and possesses an exceptionally high degree of purity. Air is only harmful when water is also present in the machine, and the action of the latter is diminished by the glycerine which is now almost universally employed for lubricating and with which water mixes in all proportions. Since glycerine is also frequently used for lubricating the compression pumps used in preparing the carbon dioxide, its occurrence as an impurity in the carbon dioxide bought in bottles is as intelligible as it is harmless.

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[1] Zeitschrift für Kälteindustrie, 1897; Journal für Gasbeleuchtung und Wasserversorgung, 1897.

[2] Zeitschrift für Kälteindustrie und Wochenschrift für Brauerei, 1897.

[3] Chemiker-Zeitung, 1895; Zeitschrift für Kälteindustrie, 1895.

Regarding the action of the impurities of the *sulphur dioxide,* there existed until recently considerable differences of opinion, which have been only partially explained by *Lange.*[1] The latter showed that pure liquid SO_2 begins to attack iron very slightly but still appreciably at +205° F., whilst the presence of water, which SO_2 absorbs to the extent of 1 per cent, lowers the temperature of attack to 158° F. As we shall see, such temperatures as these are only attained after compression in the machines, when the SO_2 is perfectly gaseous and contains no longer any trace of liquid, so that no real danger exists. In fact, after extended working of machines of this type, no chemical action on the cylinder walls or valves could be detected; *Lange* ascribes this to the action of the compressor cooling, although this could not protect the pressure-valves. Further, the fear expressed by *Venator*[2] that air and water might penetrate the vaporiser of sulphur dioxide machines and form crystallised hydrates (SO_2+nH_2O)—which would not only affect the passage of heat through the walls of the pipes, but would also enter the compressor and exert a grinding action—has never been fulfilled in practice.

From these considerations we may therefore draw the conclusion that, when sufficiently pure, *the three most important refrigerants, ammonia, carbon dioxide, and sulphur dioxide, exert no chemical action on the parts of the cooling-machine.* Their influence on lubricating materials will be considered when we come to describe the construction of stuffing-boxes.

8. Physical Properties of the Most Important Refrigerants.—Before we pass on to describe the construction and working of the machines making use of ammonia, carbon dioxide, and sulphur dioxide,[3] it will be well to consider the properties of these substances in relation to their *power of developing cold.* This appears all the more necessary since in practical circles little clearness prevails on this point, which is the subject of much dispute among the makers of the different cooling-machines and one on which the buyer is generally perplexed. Although of scarcely any practical importance, we shall consider also water, or rather water-vapour (H_2O), since, as will be shown later, it affords a convenient standard of comparison.

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[1] Zeitschrift für Kälteindustrie, 1899, p. 81 *et seq.*

[2] *Ibid.*, p. 132.

[3] Besides the mixture referred to in the preceding paragraph, of sulphur dioxide with a few per cent of carbon dioxide (so-called *Pictet's liquid*),

For ascertaining the cooling power of a substance it is important to know two magnitudes, the so-called *liquid heat* —giving the number of B.T.U. necessary to raise 1 lb. of the liquid from 32° F. to any other temperature—and the *latent heat* or *heat of vaporisation,* which tells us the number of B.T.U. required for the complete evaporation of the liquid. The latter value varies with the temperature, diminishing when this is raised, and disappears at the critical state of the substance, in which liquid and vapour can no longer be differentiated and above the temperature of which liquefaction becomes impossible. With the knowledge of these two values,

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which differs little in physical properties from pure sulphur dioxide, there is also met with occasionally *methyl ether,* which was proposed long ago by Prof. *Linde,* and *carbon bisulphide,* the inflammability of which prevents its extended use. This latter property, together with the magnitude of the compressors required, was also the cause of the early disappearance of the ether machine, in which *ethyl ether* circulated, as proposed by *Siebe.* In France machines have lately been worked with *methyl chloride* (CH_3Cl), which, according to a paper by *L. Zigliani* (Zeitschr. für die ges. Kälteindustrie, 1898 and 1899), have given good results. *Landolt and Börnstein's* Tables (second edition, Berlin, 1894) give for this compound the heat of vaporisation 174·4 B.T.U. at 32° F., and the volume 7·1 cubic feet per pound, while (according to *Regnault*) the absolute pressures (pounds per square inch) corresponding with different temperatures are:

Temperature........	−4° F.	14° F.	32° F.	50° F.	68° F.	86° F.
Pressure...........	17·1	25·3	36·6	51·6	70·9	95·4

This substance hence approaches most nearly to sulphur dioxide, but requires compressors of greater capacity, although under normal conditions no vacuum is to be feared in the machine. Since no data exist concerning the specific heat of the liquid, no further comparison can be made between methyl chloride and other cooling-agents.

which have been determined by numerous experimenters for a large number of substances at different temperatures and have been collected in tables, it is possible, for given temperatures of the cooling-agent in front of and behind the regulating-valves common to all cold-vapour machines, to obtain directly the amount of cold to be expected from unit weight (1 lb.) of the cooling-agent. This is expressed almost exactly[1] by the latent heat at the temperature of evaporation, diminished by the difference of the heats of the liquid corresponding with the two temperatures, the quantity of heat thus arrived at being that taken up in the vaporiser. From this it is immediately seen that the availability of the heat of vaporisation diminishes as the ratio of the heat of the liquid to it increases. Table I gives the values of these two magnitudes for certain temperatures.

TABLE I.[2]

Temperature, F.	Heat of Vaporisation in B.T.U. per Pound.				Liquid Heat in B.T.U. per Pound.			
	NH_3	CO_2	SO_2	H_2O	NH_3	CO_2	SO_2	H_2O
−4°	589·0	117·6	171·0	1116	−31·21	−17·19	−11·16	−36
+14°	580·2	110·7	168·2	1105	−15·89	− 9·00	− 5·69	−18
32°	569·0	99·8	164·2	1092	0	0	0	0
50°	555·5	86·0	158·9	1080	+16·51	+10·28	+ 5·90	+18
68°	539·9	66·5	152·5	1067	+33·58	+23·08	+12·03	+36
86°	521·4	27·1	144·8	1053	+51·28	+45·45	+18·34	+54
104°	500·4	.....	135·9	1042	+69·58	.....	+24·88	+72

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[1] It must be specially emphasised that this simple treatment disregards the work—generally small in amount—corresponding with the difference of pressure between the condenser and vaporiser, and hence holds only approximately.

[2] The vapour tables for ammonia and carbon dioxide are those of *Mollier*, Zeitschr. für die ges. Kälteindustrie, 1895, pp. 69 and 91; those for sulphur dioxide are taken or calculated from *Zeuner*, Technische Thermodynamik, vol. II. The same is the case with Table II. The pressures in the latter are, as must be expressly emphasised, to be taken as *absolute*, i.e., as calculated from absolute vacuum. From the *excess pressure* (calculated as starting from the atmospheric pressure) used technically, the absolute pressure differs by about 14·7 lbs. per square inch. See also foot-note, p. 7.

From this it follows, owing to the unavoidable part played by the heat of the liquid in the vaporiser, that, as a cooling-agent, water is affected least of all, ammonia and sulphur dioxide rather more but still only inconsiderably, and carbon dioxide to a very great extent. In spite of these facts we find that the apparently most suitable substance—water—is not used in compression machines, but only sulphur dioxide, carbon dioxide, and, by far the most frequently, ammonia. An explanation, though not an exact one, of this is afforded by a consideration of the pressures under which these substances exist at the above temperatures and of the volumes they occupy as absolutely dry saturated vapours. These volumes, referred to 1 lb., together with the yield of cold obtained from Table I, determine the dimensions of the compressor cylinder. They are given in Table II with the corresponding pressures.

Table II contains, like Table I, no values for CO_2 at 104°, since the critical point of this substance is passed at +88.43° F., at which the absolute pressure is 1071 lbs. per sq. inch. The fact that, beyond this condition, no change in the state of aggregation takes place—so that the heat of vaporisation disappears—early gave rise to the supposition that carbon dioxide machines became unworkable when their condenser pressure exceeded the critical or when it was not possible to lower the temperature before the regulating-valve below the critical value.

TABLE II.

Temperature, F.	Absolute Pressure in Pounds per Square Inch.				Volume of 1 Pound in Cubic Feet.			
	NH_3	CO_2	SO_2	H_2O	NH_3	CO_2	SO_2	H_2O
−4°	27·1	288·7	9·27	0·0171	10·33	0·312	8·06	15940
+14°	41·5	385·4	14·75	0·0398	6·92	0·229	5·27	7232
+32°	61·9	503·5	22·53	0·0853	4·77	0·167	3·59	3375
+50°	89·1	650·1	33·26	0·1721	3·38	0·120	2·44	1738
+68°	125·0	826·4	47·61	0·3258	2·47	0·083	1·71	941
+86°	170·8	1040	66·36	0·5902	1·83	0·048	1·22	531·0
+104°	227·7	.....	90·30	1·027	1·39	.....	0·88	314·8

This supposition was not, however, verified in practice, for it was found that these machines do not fail even in the most extreme cases, although their efficiency falls considerably below its value for normal condenser pressures. This behaviour cannot be gone into fully here, but it depends on the values of *the specific heat of carbon dioxide at constant pressure which, in this region, are very high and change rapidly with change of temperature*; this renders possible the removal in the condenser of considerably larger quantities of heat than the equivalent of the work done in the compressor and so maintains the cooling power.[1]

With the help of our two tables it becomes easy to compare the theoretical cooling effects and the compressor dimensions of machines working with different cooling-agents and at given temperatures (and pressures) in the condenser and vaporiser and before the regulating-valve. It is not at all necessary for this latter temperature to coincide with the saturation temperature in the condenser, since the cooling-agent can be cooled to the initial temperature of the cooling-water by the counter-current apparatus. In order to determine at the same time the influence of this *supercooling*, we shall take from the tables the values for two cases, namely, for a temperature in the evaporator of +14° F. and in the condenser of +68° F.; we shall suppose that in the one case the cooling-agent flows directly through the regulating-valve with this latter temperature (+68° F.), while in the second case it is previously cooled down to +50° F. We then obtain Table III, which gives us the cold obtainable from a pound of each of the substances in the two cases and also the amount of substance required to yield 100,000 B.T.U. per hour in the vaporiser at +14°; further, by multiplying by the volume of the vapour given in Table II we arrive at the stroke-volume of the compressor, i.e., the volume passed through by the piston per hour.

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[1] For details on this subject see *Mollier's* papers in the Zeitschrift für Kälteindustrie, 1895 and 1896, and that of the author on "Comparative Theory and Calculation of Compression Cooling-machines" in the same journal for 1897, where the theory is extended to these cases

Table III shows at once that, with—as is now very nearly the case practically—absolutely dry vapours, the stroke-volume in the compressor for a given cooling effect (100,000 B.T.U. per hour) is smallest for CO_2, larger with NH_3 and SO_2, while for H_2O it would lead to enormous dimensions for the compressor. Hence, just as with the cold-air machines—now given up—so also with water-vapour, tremendous frictional losses might be expected, quite apart from the danger of the

TABLE III.

Temperature in front of the regulating-valve	+68°			
Cooling-agent	NH_3	CO_2	SO_2	H_2O
Vaporiser pressure (in lbs. per sq. in.)	41·5	385·4	14·75	0·0398
Condenser pressure (in lbs. per sq. in.)	125·0	826·4	47·61	0·3258
Heat of evaporation in the vaporiser (B.T.U.)	580·2	110·7	168·2	1105
Heat imparted to the liquid (B.T.U.)	49·47	32·08	17·72	54
Cold produced per pound (B.T.U.)	530·73	78·62	150·48	1051
Amount of cooling-agent circulating for a yield of 100,000 B.T.U. per hour (in pounds)	188·4	1272	664·3	95·2
Stroke-volume (cubic feet per hour) of the compressor per 100,000 B.T.U.	1300	292	3507	688,200
Stroke-volume per hour, compared with that for CO_2 as unity	4·4	1	12·0	2350

Temperature in front of the regulating-valve	+50°			
Cooling-agent	NH_3	CO_2	SO_2	H_2O
Vaporiser pressure (in lbs. per sq. in.)	41·5	385·4	14·75	0·0398
Condenser pressure (in lbs. per sq. in.)	125·0	826·4	47·61	0·3258
Heat of evaporation in the vaporiser (B.T.U.)	580·2	110·7	168·2	1105
Heat imparted to the liquid (B. T.U.)	32·4	19·28	11·59	36
Cold produced per pound (B.T.U.)	547·8	91·42	156·61	1067
Amount of cooling-agent circulating for a yield of 100,000 B.T.U. per hour (in pounds)	182·5	109·4	638·5	93·7
Stroke-volume (cubic feet per hour) of the compressor per 100,000 B.T.U.	1264	242	3365	676,600
Stroke-volume per hour, compared with that for CO_2 as unity	5·2	1	13·8	2790

entrance of external air into the machine accompanying the extremely low pressures (high vacuum). On these grounds water-vapour is scarcely ever seriously considered as a cooling-agent for use in pure compression machines.

9. **The Indicated Amount of Work Required by Compression Cooling-machines,** which is of the greatest importance for making a practical comparison, is composed of separate items which may vary very considerably with the arrangement and use of the machine. These separate amounts are divided into those which, for given higher and lower temperatures, are determined mainly by the choice of the system, and others depending on the arrangement of the installation, especially of its pipes.

In the first group of items the principal position is occupied by the so-called *indicated compressor work,* measured by the indicator. This is not the place to calculate its magnitude with the help of thermodynamics; it may, however, be easily proved that, for a definite cooling effect and under the same conditions, its value varies for different cooling-agents. If we imagine first of all a cooling-agent the liquid volume of which may be neglected in comparison with that of the vapour, as also may the heat of the liquid entering the vaporiser compared with the heat of vaporisation—conditions which are nearly fulfilled in the case of water-vapour (see preceding paragraph),[1]—the passage through the regulating-valve will be accompanied by no appreciable loss, and the compressor work under these conditions may be regarded as a minimum to which practical working should strive to approximate.

For the example given in Table III this smallest amount of work for a cooling effect of 100,000 B.T.U. in the vaporiser will, as our earlier calculations (Chap. I, § 4) show, amount to about 4·98 H.P., so that it is of course of no importance whether supercooling takes place in front of the regulating-valve or not, since, for the cooling-agents considered, the amount of heat here escaping disappears in comparison with the heat of liquefaction. It is further assumed that the cooling-agent can pass without resistance through the valves of the compressor and the pipes of the whole installation.

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[1] This is also the ground on which water-vapour was chosen as a standard in our tables.

In order to obtain, from this minimum amount of work, the real quantity required for the different cooling-agents, we have first to take into consideration the fact that the above value was obtained on the assumption that the whole of the heat of evaporation,[1] with the exception of the heat of the liquid, is used for the production of cold. Since this is not really the case, the amount of work required must first be increased *in the ratio of the heat of evaporation to that actually used;* as is shown by a glance at Table III, supercooling in front of the regulating-valve here plays a considerable part.

The work required by the compressor may be also directly estimated if we know the *mean pressure* for every suction and direct pressure, the values of which are given by the temperatures in the vaporiser (neglecting the excess valve pressure, to be further mentioned later). If for all cooling-agents we take the same law of adiabatic compression, which approximately holds, the mean pressure for a given ratio of the *absolute condenser pressure to the vaporiser pressure* is obtained by multiplying the latter by the numbers of the second column of Table IV. In the indicator diagram this mean pressure is represented (Fig. 8) by the height of the rectangle having an area equal to

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[1] We assume, as in the preceding paragraph, that the vapour drawn from the compressor is always dry and saturated. This has been shown to be more economical than the suction of wet vapours, although the latter are more convenient for the engineer, since it is only necessary to take care that the *pressure pipes are not more than warm to the hand.* In most cases this rule is to be found among the instructions given by the makers of these machines. The extra work required with dry suction is, as may also be shown theoretically (see my paper in the Zeitschrift f. d. gesamte Kälteindustrie, 1897: *Comparative Theory and Calculation of Compression Cooling-machines*), to a great extent counterbalanced by the increased cooling action, while the apprehensions regarding *overheating* have not been fulfilled. Naturally the overheating should not be carried too far and should be as far as possible avoided for the vapour drawn out.

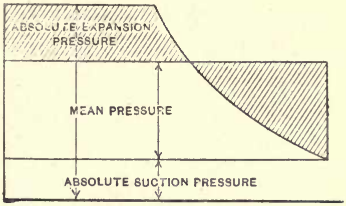

FIG. 8.

that of the diagram. The third column then gives (if at first dry saturated vapour is drawn out) the *ratio between the absolute end temperature and the initial temperature* (evaporator temperature) and so gives a measure of the overheating taking place in the machine.

By multiplying the mean pressure obtained from this table by the piston area in square inches, the *mean acting piston pressure* is obtained and this, multiplied by the length of path traversed by the piston per second (hour), gives the *work* in *foot-pounds* per second (per hour). As will be evident, the same thing is obtained if the mean pressure in pounds per square foot is multiplied by the volume described by the piston in cubic feet per second or per hour. Finally, if this value is divided by 550 ft.-lbs. per second or 1,780,000 ft.-lbs. per hour, the work in horse-power is obtained.

TABLE IV.

Ratio of			Ratio of		
Condenser Pressure to Vaporiser Pressure.	Mean Pressure to Vaporiser Pressure.	Final to Initial Temperature.	Condenser Pressure to Vaporiser Pressure.	Mean Pressure to Vaporiser Pressure.	Final to Initial Temperature.
1·0	0	1·000	4·0	1·684	1·389
1·2	0·186	1·043	4·2	1·711	1·395
1·4	0·350	1·081	4·4	1·766	1·408
1·6	0·487	1·115	4·6	1·829	1·423
1·8	0·630	1·145	4·8	1·891	1·437
2·0	0·752	1·173	5·0	1·947	1·450
2·2	0·865	1·200	5·2	2·006	1·463
2·4	0·970	1·224	5·4	2·062	1·476
2·6	1·070	1·247	5·6	2·116	1·489
2·8	1·163	1·268	5·8	2·168	1·501
3·0	1·249	1·288	6·0	2·216	1·512
3·2	1·344	1·308	7·0	2·454	1·567
3·4	1·414	1·327	8·0	2·666	1·616
3·6	1·491	1·344	9·0	2·858	1·660
3·8	1·564	1·361	10·0	3·036	1·701

Taking for our example an ammonia machine without supercooling, the absolute evaporator pressure, according to

Table II, is 41·5 lbs. per square inch, the condenser pressure 125·0 lbs. per square inch, the ratio of the two, therefore, approximately 3; according to Table IV we have to multiply the initial pressure, 41·5 lbs per square inch, by 1·249, which gives the mean pressure as 51·88 lbs. per square inch or 7471 lbs. per square foot. Since, according to Table II, 1 lb. of ammonia occupies (at +14° F.) a volume of 6·92 cub. ft., and, according to Table III, 188·4 lbs. of ammonia circulate per hour, the volume passed through per hour is $6 \cdot 92 \times 188 \cdot 4 =$ 1304 cub. ft., and the work done 9,741,000 ft.-lbs. per hour or $9{,}741{,}000 : 1{,}980{,}000 = 4 \cdot 92$ horse-power. The ratio which this number bears to the minimum work required, namely, 4·98 horse-power, is roughly 1·1, which is in good agreement with the ratio between the heat of evaporation and the available cooling effect given in Table V (second line).

Finally, the end temperature of the compression corresponding with an initial temperature of +14° F. (474° absolute) is given by $474 \times 1 \cdot 288 = 610°$ absolute or 150° F.

If, now, we allow for the fact that the resistance offered by the valves of the compressor absorbs, according to the number of rotations and size of the machine, from 5 to 10 per cent—or, on an average, 7·5 per cent—of the compressor work, we obtain from Table V the indicated work for the example given in Table III.

On comparing these last values with reliable experimental results of good machines, they are always found to be too favourable. This is owing to the influence of *internal irregularities* caused by the impossibility of making valves and pistons which fit absolutely tight. These irregularities, which are of course inconsiderable in new machines, introduce a disturbing action in all systems as time goes on; during the compression period part of the substance passes back through the badly fitting pressure-valve and reaches the suction space partly through the suction-valve and partly past the piston, while during the suction period some of the substance creeps into the cylinder and takes up part of its suction volume. This

TABLE V.

Temperature in front of the regulating-valve	+68° F.			
Cooling-agent	NH_3	CO_2	SO_2	H_2O
Minimum horse-power required per 100,000 B.T.U.	4·98	4·98	4·98	4·98
Ratio of the heat of vaporisation to the available cooling action	1·093	1·408	1·118	1·051
Increase owing to the resistance of the valves	1·075	1·075	1·075	1·075
Ratio of the total indicated work to the minimum	1·175	1·513	1·202	1·130
Total indicated horse-power per 100,000 B.T.U.	5·85	7·53	5·99	5·63
Cooling effect per hour per indicated horse-power in B.T.U.	17,100	13,300	16,700	17,800

Temperature in front of the regulating-valve	+50° F.			
Cooling-agent	NH_3	CO_2	SO_2	H_2O
Minimum horse-power required per 100,000 B.T.U.	4·98	4·98	4·98	4·98
Ratio of the heat of vaporisation to the available cooling action	1·059	1·211	1·074	1·033
Increase owing to the resistance of the valves	1·075	1·075	1·075	1·075
Ratio of the total indicated work to the minimum	1·138	1·302	1·155	1·110
Total indicated horse-power per 100,000 B.T.U.	5·67	6·48	5·75	5·53
Cooling effect per hour per indicated horse-power in B.T.U.	17,600	15,400	17,400	18,100

latter portion has had the work of compression done upon it without being available for the production of cold. The work lost in this way may be very considerabe (10 to 20 per cent), and is dependent not only on the original construction and subsequent upkeep of the machine, but also on its number of revolutions.

In spite of the fact that these losses cannot be calculated, Table V, taken in conjunction with Table III, allows us to compare the different cooling-agents. It is seen that—apart from water-vapour, which is only considered for the sake of completeness—the work required for a machine working at the two temperatures taken is least for ammonia and only slightly greater for sulphur dioxide, whilst with carbon dioxide far less favourable results are obtained. The difference is very considerable when working *without supercooling,* which must hence be regarded as absolutely necessary if cold cooling-water, even in small quantity, is obtainable. The lower efficiency of these machines compared with the other two systems has little importance under normal conditions, since the internal loss, due to the looseness of the valves, etc., in carbon dioxide compressors with leather-ringed pistons is relatively less than in other machines with metallic piston-packing.

The supercooling itself is best effected by arranging a special *liquid-cooler* behind the condenser. It is composed, like the latter, of tubes and has about $\frac{1}{10}$ to $\frac{1}{3}$ of the cooling surface of the condenser; through the tubes passes the liquid cooling-agent from the condenser, whilst round the outside and in the opposite direction fresh cooling-water flows. The latter then goes into the condenser to be used again.

As regards Table V it must be remarked that the comparative numbers obtained are *not to be considered as holding generally* or extended to other condenser and evaporator temperatures. The relation between ammonia and sulphur dioxide does not, indeed, change much under other conditions, but with rise of the condensation temperature the efficiency of the carbon dioxide machine diminishes considerably, so that, although it is capable of competing under normal circumstances, only exceptionally could it do so with a high temperature (e.g., +77° to +86°) in front of the regulating-valve.

10. Total Work Required in Compression Cooling-machines.—The work required by a compression cooling-machine is far from being exhausted by the indicated compressor work, and it is therefore inadmissible to compare the different systems with reference only to *this* work, since for the possessor of a cooling installation the *total work* required is alone of importance. This includes, e.g., in a *brewery cooling installation,* besides the above-obtained work, the following items, which are indeed almost independent of the choice of system and depend mainly on the general arrangement of the whole installation:

(*a*) *Frictional work of the machine,* which is usually, but not quite accurately, identified with the difference between the indicated work at the motor and that at the compressor.[1] Its magnitude varies with the size of the installation from 10 to 20 per cent of the indicated compressor work.

(*b*) *Work for driving the stirring-apparatus* in the condenser and evaporator, in the case where the former is cooled by water and the cold produced in the evaporator transferred to a salt solution, which then passes into pipes and abstracts heat from the spaces or objects to be cooled.

Under normal conditions of working this work should not amount to more than 2–3 per cent of the total work.

(*c*) *The work of the cooling-water pumps* only becomes appreciable if the source of the water (e.g., the well) is far removed from the machine and necessitates the use of a pipe with many bends. The latter occurs when the condensers of the cooling-plant are not placed in the engine-room but on the roof of the building, so as to make use of evaporation. The hydraulic resistance then becomes very appreciable. In general, we shall not go far wrong if we calculate this pumping work simply from the weight of water used per hour and the total height it is raised through, and then multiply this amount by about two to allow for the resistance to motion. This multiplier must be increased if the water has to be drawn from a deep boring with the help of a so-called *Mammoth pump,* which is an air-jet apparatus. Such a pump requires to be worked by an air-compressor, which is best driven, independently of the machine, by a separate steam-cylinder. Experiments[2] have shown that with these pumps the ratio of the useful work (measured by the amount of water raised through a certain height) to the indicated compressor work may be expected to be from 30 to 40 per cent. And although this efficiency is further lowered to 25–30 per cent by the ratio of the indicated compressor work to the steam work, besides which the steam cannot be used to great advantage in such small, direct-acting pumps, yet the latter must be regarded as a very satisfactory and reliable makeshift.

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[1] This is only accurate if the following amounts of work have not, as is usually the case, to be furnished by the driving engine.

[2] *Josse,* Versuche mit Mammuthpumpen (air-pressure water-raisers), Zeitschrift d. Ver. d. Ingenieure, 1898.

(*d*) *The work of the brine-pumps* depends in a very high degree on the arrangement of the network of pipes in the cold rooms (cellars) and, together with the compressor work, is of main importance. If the brine is driven with a high velocity through narrow pipes (or, what is the same thing, through a number of bent pipes arranged one behind the other), the pumps may easily consume as much as 30 per cent of the total driving energy, whilst, with a proper disposition of the cooling-pipes (partial parallel arrangement of the separate tubes) and with a large sectional area, scarcely the half of this amount is required. Owing to the great difficulty of estimating it, it is best to take the maximum value for this work when calculating the power of the driving motor.

(*e*) *The work of the pumps which circulate fresh water* for the attemperators of the fermenting-tuns need not be taken into account, since it has but a small value compared with the other items and is rarely required at the same time as the latter. In most breweries the fresh water is collected from the attemperators in a reservoir, whence it is transferred to the fresh water cooler.

(*f*) *The work required for driving the ice-generator* seldom forms an integral part of the total work, since the generator is frequently combined with the evaporator (brine-cooler), the brine being thus cooled to a lower temperature than is required by the remainder of the installation and the compressor work slightly raised. Since the cooling requirements of a brewery naturally vary with the time of day (owing to the cooler night-time), the production of ice may be used with advantage to regulate the working and the temperature. To this end, the whole of the pipes are often—even when the ice required only absorbs a fraction of the total cold generated—removed into the generator, which then becomes capable of taking up and storing the excess of cold.

(*g*) *The work of transmission* can scarcely be estimated beforehand, as it depends mainly on suitable arrangement and keeping in order (lubrication).

Of the above-mentioned quantities of work, the items (*b*), (*c*), and (*d*) are further noteworthy, since *in their destruction they are transformed into heat*[1] and thus, on the one hand, cause heating of the cooling-water and, on the other, uselessly absorb in the brine a portion of the cold generated. This is especially the case with item (*d*), which may easily diminish the amount of cold generated by 10 per cent or even more. A further diminution occurs owing to *radiation* of atmospheric heat into the evaporator, this being more important in small plants than in large ones, on account of the relatively greater area. This effect may be taken as 5 per cent for a large plant and about 10 per cent for a small one.

In the case when air has to be cooled in a special apparatus, dried and passed through ventilators into the room to be kept cool, the cold generated is impaired by the *ventilator work* necessary to overcome the resistance to motion, so that as much care must be exercised in the arrangement of the air-passages as in that of the brine-pipes.

Table VI indicates, for the various cooling-agents, NH_3, CO_2, and SO_2, the influence of the different items of work on the effective driving work. It is based on the values of Table IV and supposes a theoretical generation of cold of 100,000 B.T.U. By irregularities of valves, etc., in the machine this is lowered by 10 per cent, that is to 90,000 B.T.U. per hour, for SO_2 and NH_3 machines, and by 5 per cent, that is to 95,000 B.T.U. per hour, for CO_2, without the indicated work of the compressor changing.

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[1] The transformation of 1 horse-power yields roughly 2550 B.T.U. per hour.

TABLE VI.

Temperature before the regulating-valve	+68° F.		
Cooling-agent	NH_3	CO_2	SO_2
Total indicated H.P. (from Table V)	5·85	7·53	5·99
Friction of the compressor and driving-machine in H.P.	0·90	1·10	0·90
H.P. for driving stirring-apparatus	0·3	0·3	0·3
" " " cooling-water pumps	0·45	0·45	0·45
" " " brine-pumps	2·2	2·2	2·2
Energy of transmission in H.P.	0·6	0·6	0·6
Effective total driving H.P.	10·30	12·18	10·44
Cooling loss through stirring (B.T.U.)	765	765	765
" " " brine-pumps (B.T.U.)	5610	5610	5610
" " " radiation (B.T.U.)	4500	4500	4500
Residual useful cooling effect (B.T.U.)	79,125	84,125	79,125
Cold generated per effective H.P. (B.T.U.)	7682	6908	7578

Temperature before the regulating-valve	+50° F.		
Cooling-agent	NH_3	CO_2	SO_2
Total indicated H.P. (from Table V)	5·67	6·48	5·75
Friction of the compressor and driving-machine in H.P.	0·85	0·95	0·85
H.P. for driving stirring-apparatus	0·3	0·3	0·3
" " " cooling-water pumps	0·45	0·45	0·45
" " " brine-pumps	2·2	2·2	2·2
Energy of transmission in H.P.	0·6	0·6	0·6
Effective total driving H.P.	10·07	10·98	10·15
Cooling loss through stirring (B.T.U.)	765	765	765
" " " brine-pumps (B.T.U.)	5610	5610	5610
" " " radiation (B.T.U.)	4500	4500	4500
Residual useful cooling effect (B.T.U.)	79,125	84,125	79,125
Cold generated per effective H.P. (B.T.U.)	7828	7662	7796

From this table, the values of which are for normal conditions of working and for equally good arrangement and construction, it is seen that *for the various cooling systems no appreciable difference exists in the work required for the same amount of cold delivered at the place where it is to be applied,* and that these are hence to be regarded as of equal value. If greater differences are occasionally met with in practice, they must be ascribed either to faulty working, or to unsuitable arrangement, or to wrong or negligent treatment of the machines.

11. Energy Required by Absorption Cooling-machines.—We have already seen (§ 6) that the energy required by absorption machines, unlike that of compression machines, is supplied mainly in the form of heat. The estimation of this quantity of heat beforehand is only possible when we know exactly the physical properties of solutions of ammonia in water. As this is only very incompletely the case, we are often driven to assumptions and can hence only expect an approximate result. First of all one sees that the pressures in the machine before and behind the pump determine the pressures in the evaporator and condenser and the corresponding temperatures (see Table II). Further, the amount of ammonia circulating in the machine per hour is given by Table I and Table III. This ammonia combines in the absorber with water with evolution of roughly 900 B.T.U. per pound of ammonia (determined by v. Strombeck at 62·5° F.) and forms a more or less saturated solution. The saturation of this solution depends not only on the temperature in the absorber, but also on the pressure in the evaporator, while the degree of saturation of the solution freed from gas in the heater is determined by the temperature of the heating steam and the condenser pressure. This dependence on the pressure is known exactly for only a very short interval, but it increases with the pressure and for dilute solutions in approximately the same ratio. The variation of the solubility with the temperature is given in Table VII, which contains also the specific gravities of the solutions.[1] The pressure is here taken as that of the atmosphere (29·9 inches of mercury).

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[1] See *Landolt and Börnstein*, Phys. Chem. Tabellen, Second Edition, p. 221. The data given in this work extend from the sp. gr. 0.88 to 0.97; the other values were calculated by means of the specific gravities of liquid ammonia from the data of *Lange* (Zeitschr. f. Kälteindustrie, 1898, p. 45). The table therefore is not so accurate below +104° F. The data on the solubility of ammonia below 32° are those of *Mallet* (American Chemical Journal, 1897, **19**, p 804).

In the heat-change going on in the interchanger the specific heats of the two solutions meeting there come into play. We have no data on this head, but since the specific heat of liquid ammonia has just about the same value as that of water, we shall use this value also for ammonia solutions of various concentrations. Further, no measurements have been recorded of the amount of heat required for expelling ammonia from its solutions at high temperatures and pressures; we are able, however, from the balance of heat, to determine this quantity, which is necessary for a review of the whole process.

We now pass to the calculations for an absorption machine with a cold production of 100,000 B.T.U. per hour at an evaporator temperature of +14°, the temperature in both the condenser and the absorber being in one case +68°, and in another +86°, and no supercooling taking place. The corresponding absolute pressures are, in the evaporator 41·5 lbs. per square inch and in the condenser, 125·0 or 170·8 lbs. per square inch.

Table I shows that 1 lb. of ammonia carries into the evaporator a liquid heat of 49·5 B.T.U. at a condenser temperature of +68° F. (67·2 B.T.U at +86° F.), so that it must there take up 530·7 B.T.U. (or 513) for complete evaporation.

TABLE VII.

Temperature, Degrees F.	Pounds of Ammonia Dissolved by 1 lb. of Water.	Specific Gravity, Pounds per Gallon.	Temperature, Degrees F.	Pounds of Ammonia Dissolved by 1 lb. of Water.	Specific Gravity, Pounds per Gallon.
−40	2·94	7·3	+104	0·34	8·8
−22	2·78	7·4	+122	0·28	9·0
− 4	1·77	7·5	+140	0·24	9·1
+14	1·11	7·8	+158	0·19	9·3
+32	0·90	7·9	+176	0·15	9·4
+50	0·68	8·0	+194	0·11	9·5
+68	0·52	8·2	+212	0·07	9·7
+86	0·41	8·4			

Hence for the generation of 100,000 B.T.U. per hour 188 (195) lbs. of ammonia must circulate. In the absorber a lower pressure prevails than in the evaporator—since the vapour passes out from the latter of itself—and this we estimate at roughly 28 lbs. per square inch. If we had only atmospheric pressure, 1 lb. of the solution would, according to Table VII, absorb 0·52 (0·41) lb. of ammonia in order to become saturated. This saturation is, however, never reached, whilst, on the other hand, absolutely ammonia-free water never enters the absorber. We shall therefore assume that the diminution in the absorptive power thus produced is counterbalanced by the higher pressure, so that we obtain for 1 lb. of ammonia 1·93 (2·44) lbs. of water, giving 2·93 (3·44) lbs. of solution. The heat of absorption, 900 B.T.U., has to be removed during the combination and, indeed, diminished by the amount serving for the heating of the ammonia vapour from the temperature of the evaporator to that of the absorber. The specific heat of NH_3 being 0·54, we have a deduction of $0{\cdot}54\times(68-14)=29{\cdot}2$ (or $0{\cdot}54\times72=38{\cdot}9$) B.T.U. per pound, so that altogether 871 (861) B.T.U. per pound of ammonia must be removed from the absorber per hour; for the whole quantity of 188 (195) lbs. of ammonia the total heat to be withdrawn is hence 163,800 (167,900) B.T.U. per hour.

Further, the work of the pump is that required to raise the hourly amount just obtained, namely, $2{\cdot}93\times188=551$ lbs. ($3{\cdot}44\times195=671$ lbs.) from a pressure of 41·5 lbs. per square inch to 125·0 (or 170·8) lbs. per square inch; i.e., in round numbers, 44,360 (or 86,760) ft.-lbs. per hour or 0·22 (0·44) horse-power. This extraordinarily small amount of work corresponds with an hourly heat-equivalent of 57 (or 112) B.T.U., which can only play a subordinate part in the balance of heat.

The boiler is heated with steam at 302° F., which experience shows to be capable of expelling the ammonia. From this there escape the ammonia vapour to the condenser and the weak solution at a temperature[1] of 302° to the interchanger. The latter solution, amounting to 1·93 (or 2·44) lbs. per pound of ammonia, can be cooled in the interchanger to very near the absorber temperature, giving up $1{\cdot}93(302-68)=452$ [or

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[1] In most cases (see later) the ascending hot gases meet the incoming cold strong solution, whereby an exchange of heat takes place in the upper part of the boiler; this does not in any way affect the calculation.

2·44(302 − 86) = 527] B.T.U. per pound of ammonia. By this heat the opposite stream of strong solution is to be heated. Since the latter weighs more than the absorbed ammonia, namely, 2·93 (or 3·44) lbs. per one of ammonia, it will require, to heat it to the boiler temperature, 2·93 (302 − 68) = 686 [or 3·44(302 − 86) = 743] B.T.U. The difference of 686 − 452 = 234 [or 743 − 527 = 216] B.T.U. per pound of ammonia has to be furnished by the hot steam; for the quantity of ammonia under consideration this heat amounts to 43,980 (or 42,120) B.T.U. per hour.

The heat necessary for the expulsion of the ammonia is not yet known, but we can calculate that yielded by the condenser. This is divided into two parts, that given out in the cooling of the vapour (*heat of superheating*) to the condenser temperature of 68° or 86°, and the heat of liquefaction. The specific heat of ammonia vapour being 0·54, the first of these amounts will be, for a temperature of 68° in the condenser, 0·54(302 − 68) = 126 B.T.U. per pound per hour, or, altogether, 126 × 188 = 23,690 B.T.U.; or for a condenser temperature of 86°, 0·54(302 − 86) = 117 B.T.U., or a total of 117 × 195 = 22,810 B.T.U. The heat given out by the liquefaction is, in round numbers, 540 (or 522) B.T.U. per pound, or, for the whole amount, 101,500 (or 101,800) B.T.U. per hour.

We are now in a position to make up the *balance-sheet of the heat* and so obtain the *total heat* required for working. The different items for condenser temperatures of both 68° and 86° are given in the table on the opposite page.

The sum of the last three quantities, namely, 301,811 or 298,516 B.T.U., *must be regarded as the addition of energy corresponding with the work of the compression machine.*

In a comparison of these numbers with those obtained above for compression machines, it must be borne in mind that they are only rough approximations and that they have been calculated without taking account of the energy losses in the machine. These losses, which can only be estimated, raise the energy to be supplied to at least 330,000 B.T.U.

It will further be noted that, *in absorption machines, the*

	+68°	+86°
1. Heat of absorption	163,800 B.T.U.	167,900 B.T.U.
2. " " superheating	23,690 "	22,810 "
3. " " liquefaction	101,500 "	101,800 "
Hence total heat to be withdrawn	288,990 "	292,510 "
4 Cold generated	100,000 "	100,000 "
5· Equivalent of the pump-work	57 "	112 "
6· Heating in boiler	43,980 "	42,120 "
7·. Heat of expulsion[1]	257,774 "	256,284 "

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[1] This corresponds with a value of 769–772 B.T.U. for the expulsion of 1 lb. of ammonia at 302°. It is seen that, by the above method of calculation, we can obtain the amount of heat required at any temperature other than 62.6° F. for which it was experimentally determined. A more exact theoretical development, based on thermochemical considerations, was given by me under the title "The Method of Working and Calculation of Ammonia Absorption Machines," in the Zeitschr. für die gesamte Kälteindustrie, 1899. Since the final results there arrived at differ but little from those given in the text (p. 44), it will suffice to give this reference for those readers acquainted with thermodynamics.

energy required does not increase with the temperature of liquefaction, as occurs in compression machines; this also explains the fact that in hot districts this class of machine has persisted. To the extraordinarily high energy requirements correspond also the withdrawal of a large amount of heat and the use of *large quantities of cooling-water.*

The energy used up in these machines is not, however, very much greater than that required by compression machines, if the motive power of the latter is supplied by a steam-engine. Taking 1170 B.T.U. as the amount of heat to be supplied to the boiler per pound of steam, our absorption machine requires about 282 lbs. of steam per hour for a cooling effect of 100,000 B.T.U.; on the basis of 20 lbs. of steam used per horse-power, this corresponds with 14 horse-power.

The conditions would be more favourable if the engine could be driven by the waste steam from a large steam-engine, since this would at the same time work the small pumps, which, when driven direct from the boiler, necessitate a large expenditure of steam.

The total energy required must also take into account the amounts used up by certain accessories, such as stirring-apparatus and brine-pumps, and these may be given the same values as in compression machines, except that the larger quantity of cooling-water used necessitates a correspondingly larger expenditure of energy in its supply.

12. Driving-engines for Cooling-plants.—The choice of the driving-engine for a cooling-plant often receives but little consideration, although it is usual to treat economically the cooling machinery itself. Consequently it happens that not only are good cooling-machines frequently made to suffer by faulty combination with their driving-engines, but very often, also, a really bad steam-engine introduces disturbances into the working, or at least causes an immoderately great expenditure of coal.

In the choice of such a motor the first question to be decided is whether it is to be used solely for the cooling-plant or whether it is to serve other purposes also. The first case is only exceptionally met with in large plants, while in medium-sized or small breweries, and also in slaughter-houses, the driving-engine has to supply power for the whole of the work required. Such combined working, which concerns mainly the steam-boiler—in so far as this has to supply steam for other purposes, —renders regularity in the working of the cooling-machine very difficult of attainment, and necessitates also, for small plants, good and automatic regulation. It is practically immateria whether the steam-engine is coupled up directly with the compressor or whether the latter is driven by a transmitter.

Such a transmission compressor is shown in plan in Fig. 9, while Fig. 10 represents the normal method of coupling-up with a steam-engine. Since in the latter case the whole of the compressor work is transmitted by the shaft, the latter must be made very strong. That the greatest care must be taken in the erection, especially in the parallel arrangement of the axes of the two cylinders, need hardly be mentioned. The same is the case

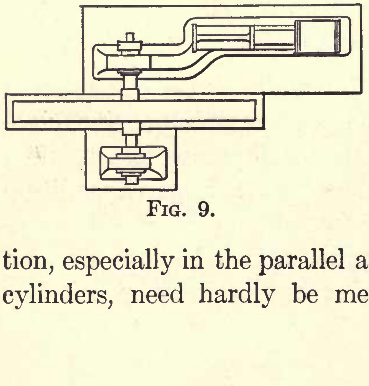
FIG. 9.

for the coupling-up of the oppositely arranged *double compressors* with the motor (Fig. 11), often found in large cooling-

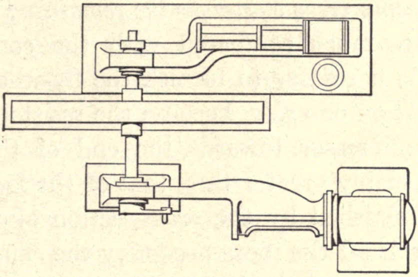

FIG. 10.

plants. If the engine works with double expansion, as is to be recommended for large plants on account of the saving in uel, the two steam-cylinders are generally arranged one directly behind the other (*tandem* system). Only rarely are the two cylinders placed opposite to one another (*vis-à-vis* system) like the compressors in Fig. 11, although less space is taken up in this way.

With direct coupling the excess of work is transmitted by the fly-wheel made in the form of a belt- or rope-pulley, or

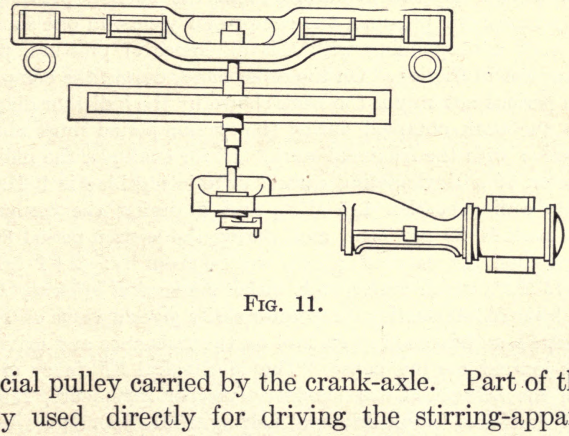

FIG. 11.

by a special pulley carried by the crank-axle. Part of this work is mostly used directly for driving the stirring-apparatus of the condenser and evaporator, or for working the travelling cranes and tipping-apparatus of ice-generators. In any case, *direct coupling* may be advantageously employed *if the work of driving the compressor outweighs the remaining work,* or if the latter fluctuates much compared with the compressor work. Very often it is necessary to furnish the fly-wheel with a *compensating weight* on one side, because the resistance of the back side of the compressor towards the end of the compression period is appreciably greater than that of the front side, where the area is diminished by the cross-section of the piston-rod. This balancing is all the more necessary the smaller the diameter of the cylinder and the larger the piston-rod, and is found most frequently in carbon dioxide compressors. The most unsuitable are small, *single-acting engines,* mostly of vertical construction,[1] the frictional losses of which are very great. For these the use of a balancing-weight on the fly-wheel is almost always necessary to obtain smooth running, even when they are driven by transmission. With large, double-acting machines driving indirectly, the energy stored up in the transmitter generally suffices to overcome inequality in the resistance. This is most evident on *starting* the machine, especially if the suction is not stopped (cut-off valve), and in direct-coupled non-balanced compressors causes many difficulties. This difficulty is best treated by subsequently bringing the weight down, the occurrence of greatest resistance in the cylinder determining its position with reference to the compressor-crank. At the same moment the balancing-weight must exert its greatest turning moment on the axis; so that it will, e.g., with horizontal carbon dioxide machines, in which the maximum pressure on the piston occurs on an average at the middle of the stroke, have a lead of 90° on the compressor-crank; in vertical machines the lead will be 180°. In settling the size of this weight it must not be forgotten that during half the revolution it is destroying work and so increasing the resistance of the other side of the compressor. It is therefore best simply to compensate for the difference of work on the two sides of the piston by means of the weight, the size of which is then obtained directly from the given fly-wheel diameter.

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[1] If, as always happens in normal cases, the suction pressure is greater than the atmospheric pressure, the compressor yields, in one suction period, positive work corresponding with this difference of pressure, part of this work being lost in friction. On the other hand, during the compression and expulsion periods not only is the work shown by the indicator diagram necessary, but the work obtained during the suction period must also be given back together with the frictional work. If, for example, the indicated compressor work of a single-acting carbon dioxide machine is 2 H.P. and the work done during suction 1·5 H.P., and if during the compression and expulsion periods 20 per cent, and during the suction period 30 per cent, is lost by friction, the *real driving work* will be given by: $(2+1\cdot5)1\cdot2-1\cdot5\times 0\cdot70=3\cdot15$ H.P., which corresponds with a *mechanical efficiency of* $2/3\cdot15=0\cdot635$; a large, double-acting compressor easily gives a value of 0·85 or 0·9.

For details on frictional losses and on the resistance and driving force at the crank-journal see my paper, "The Mechanical Efficiency of Piston-engines," in the Zeitschrift des Vereins deutscher Ingenieure, 1894, and my book, "Dynamics of Crank-driving," Leipzig, 1901 (§ 4).

The same end is attained by the *crossing of the cranks* of the compressor and driving-engine, inasmuch as both act on the same axis but at opposite ends of it. Having regard to the common one-sided (for vertical forces directed downwards) guides of the compressor, the latter and therefore also the steam-engine are made to revolve left-handedly. With this arrangement it has been found best to allow the steam-engine crank to have a lead of 120–140° from the compressor-crank, but for double compressors with one crank it is usual to have the steam-engine crank 45–60° in front of one of the others in whichever direction the machine revolves.

For the exact determination of this angle and of the position of the balancing weight we may with advantage use the *graphic method*, which, if we assume a certain degree of irregularity, gives also the total fly-wheel weight at once.

For the uniform working of the machine the arrangement, favoured in France and England, of placing the compressor- and motor-pistons one directly behind the other (see Fig. 12) is very bad, since then the greatest steam pressure always corresponds with the smallest back pressure in the compressor and *vice versa.* The fly-wheel must consequently take up the whole of the excess work at the beginning of the stroke and give it back at the end. All the same, this arrangement would be worthy of notice on account of its adaptability to long, narrow engine-rooms, if only it were applicable to the relatively large sulphur dioxide compressors.

Characteristic of all motors coupled up directly with cooling-machines is their relatively small *number of revolutions* or piston velocity. The object of this is to obtain the best velocity for the compressor-piston with which the valves act most regularly and the unavoidable internal leakage is kept within allowable limits. By increasing the number of revolutions the internal leakage diminishes in relation to the cold produced per hour, but the valves do not function so exactly and fail more readily, whilst when the speed is lower, and also when the valves act more perfectly, the losses due to creeping of the gas from the pressure-pipe into the compressor and thence into the suction-pipe may be very considerable.

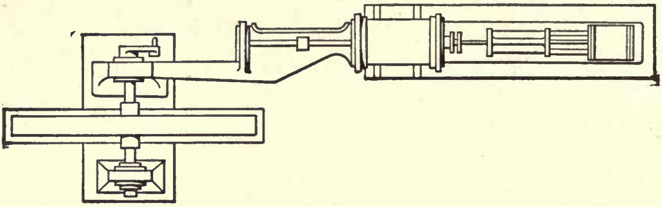

FIG. 12.

In general the most favourable number of revolutions for the compressor is much lower than that for the steam-engine, and it requires years of experience to obtain the right speed at once. This problem is somewhat simplified by the fact that the steam-engine, as already mentioned, must generally be made considerably stronger than would suffice for the compressor alone and consequently works at a speed more favourable to the compressor. But in no case can one expect to increase appreciably and permanently the yield of an existing cooling-plant by raising its speed above the normal value.[1] The result of so doing has been found to be rapid wearing-out of all the moving parts and a considerable increase in the work required.

If the compressor is driven by a quick-running *gas-motor*, a transmitter is necessary; this is also the case when an *electromotor* is employed, as it is in rare instances. When these latter are used with small plants, good results are obtained with direct coupling by means of an *endless screw*, as with careful working the frictional losses are considerably smaller than was previously assumed.

With *absorption machines* the driving does not play so important a part, as it is only a question—apart from the arrangements necessary for the moving of the brine and cooling-water—of a small pump. In most cases it will be necessary to connect this pump and also the accessory apparatus to a transmitter; otherwise it would be advisable, in order to save room and attention, to employ a small, direct-acting steam-pump without a fly-wheel, all the more so because we need not be extremely economical with fuel here.

13. General Arrangement of the Cooling-plant.—If the question of the driving of the cooling-plant is settled and a particular system of cooling—according to the requirements and the available space—decided on, the arrangement becomes an easy matter. As an example of a *compression plant*, the disposition of a Linde machine with a directly coupled steam-engine B and the accessories is shown in Figs. 13 and 14. It is seen that the compressor A with its driving-engine occupies about the centre of the engine-room, in the corners of which cylindrical iron vessels containing the coils of the condenser K and of the evaporator V find a place. Between these two and on the wall is the regulating-valve R arranged at such a height that it is readily accessible. Above the regulating-valve are generally found the manometers for observing the pressures in the two main parts of the apparatus. The stirring-apparatus consists of boards slowly rotating inside the coil round a vertical axis, which is driven by means of a pair of bevel wheels and pulley actuated by gearing (not shown in the sketch) from the wall or roof. This latter is connected by a belt with the

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[1] The only limited heat conductivity of the walls of the condenser- and evaporator-tubes here comes into play, and into this we shall go further later on (Chap. IV).

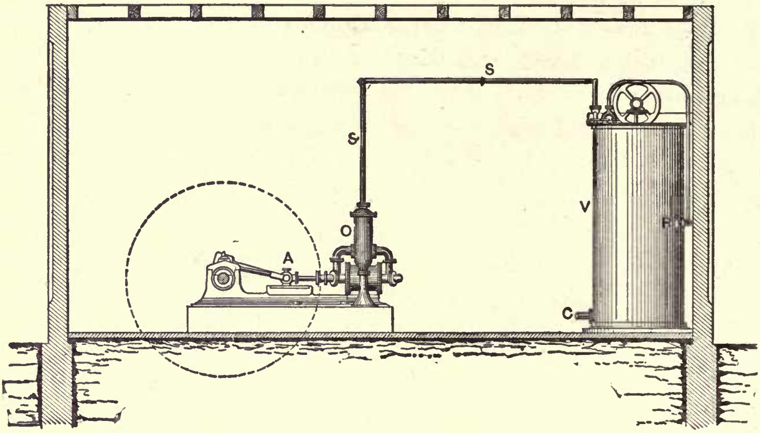


Fig. 13.

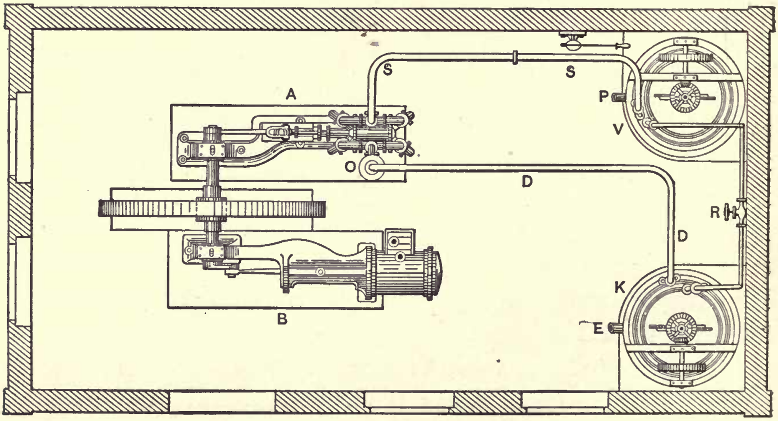


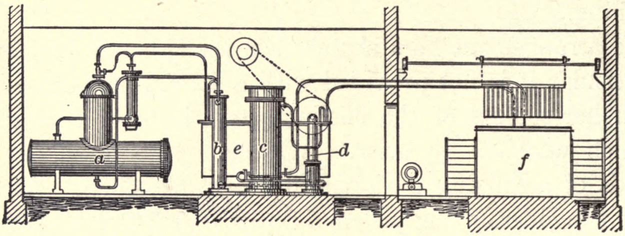
Fig. 14.

fly-wheel of the compressor. At *C* is seen the end of the pipe by which the brine proceeds to the evaporator and at *E* the entry of the cooling-water into the condenser. Beside the compressoris the oil-separator *O* inserted in the pressure-pipe *DD*, the latter and also the suction-pipe *SS* hanging near the ceiling.

The arrangement here sketched is the simplest imaginable; in particular it is not provided with a special arrangement for the further cooling of the cooling-agent, into the construction of which we shall go later. We shall also learn of other methods of arranging the apparatus just sketched. With the smaller machines having yields of up to 16,000 B.T.U. per hour, the compressor-frame is generally combined with the condenser, which is cylindrical when the piston moves vertically or rectangular when the motion of the piston is horizontal. In the latter case, the compressor, which is often (e.g., in ships) combined with the driving steam-engine, lies directly on the condenser vessel. Such crowding together is of course only justified by convenience, since not only is the necessary supervision of the plant hindered, but the radiation of heat from the steam-cylinder influences the action of the neighbouring parts of the cooling machinery.

The arrangement of an *absorption machine* is shown in the sketches given in Figs. 15 and 16, which represent an ice-making plant by *Halle's* Union (formerly *Vaas & Littmann*). The heater *a* is a horizontal boiler with internal steam-heating and a tower-like dome, in which are arranged plates one above the other. In this way opportunity is afforded the ammonia of separating any water it carries with it, i.e., of becoming rectified. Besides the dome there is (generally at the wall) a simple water-separator with a water-pipe leading to the heater and through this the ammonia vapours have to pass before they reach the condenser *e*. In order to apply a counter-current of cooling-water to the hot vapours, the condenser consists of a series of compartments, in each of which lies a number of horizontal ammonia pipes, and which are traversed one after the other by the cooling-water. From the condenser the liquefied ammonia passes through the regulating-valve (not shown) into the evaporator *f*, arranged as an ice-generator, and thence into the absorber *c*, where it unites with the weak

Fig. 15.



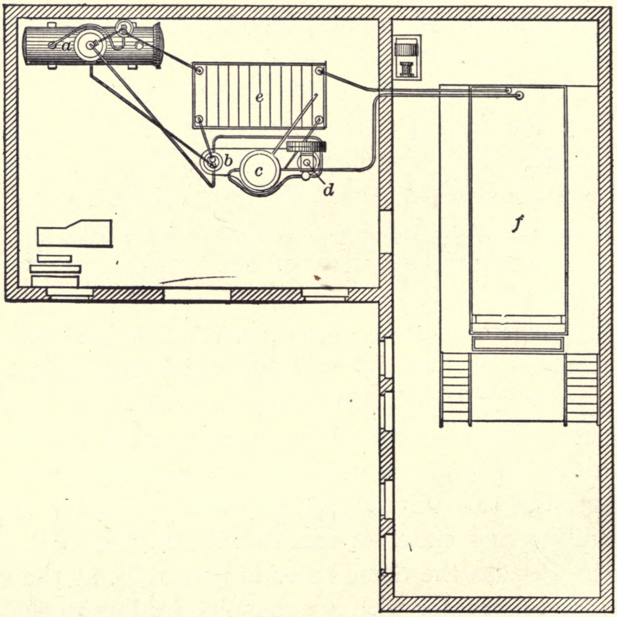


Fig. 16.

solution flowing from the heater through the interchanger *b*. The vertical pump *d* then drives the strong solution back through the interchanger *b* into the condenser, where the cycle begins again.

14. Description of Cooling-machines According to their Practical Effect.—As we saw in § 9, the yield of a compression cooling-machine depends, for given evaporator and condenser temperatures, on the stroke-volume per hour of the compressor, i.e., on the dimensions of the latter and its speed. These numbers are in general not published by the makers, although the owner of a plant can obtain them for himself. No difficulty would be found in deriving them also, with the help of certain assumptions, from our earlier considerations, just as we did in the case of the work used up. Such results would, however, possess no general value, owing to the variation in the assumptions, which depend mainly on the relation between the diameter of the cylinder and the stroke of the piston. On the other hand, it appears advisable to become acquainted here with the descriptions under which machines are generally sold. The accompanying table, which applies with but slight deviations to ice-machines from all makers except the Linde Company, gives information on this head. The description of the cooling-machines is given in roman figures, between which others have been inserted in the course of time. The table contains further the cooling effect of the machine in thousands of B.T.U. per hour for both fresh water cooling (50° to 34°) and brine cooling (28·5° to 23°), and also the usual ice production per hour with a brine temperature of from 23° to 21°. Finally is given the daily ice replacement in hundredweights, which gives—according to the method adopted in America and, in some cases, in Germany—the quantity of ice, the melting of which would be replaced by the cooling effect of the machine. As is immediately seen, this is by no means the same as the daily ice production of the plant, but is always much greater, owing, on the one hand, to the relatively high melting-point of natural ice (32°) and, on the other, to its variable and generally irregular consistency. This explains also the pretty considerable deviations in the ice-replacement data.

For the ice production, the Linde Ice-machine Company uses also the description numbers given in Table VIII, only with the difference that the numbers VI*a* and VII are replaced by VII and VIII. For cooling-machine working, however, this firm numbers according to the scheme given in Table IX.

TABLE VIII.

Model number	I	II	III	III*a*	IV
Cooling effect in thousands of B.T.U. per hour:					
From 50° to 34°	13·1	27·8	63·5	119·0	167
" 28.5° to 23°	10	20	52	99	139
Ice production in pounds per hour	44	88	220	386	550
Ice replacement per day in cwts	15	30	80	140	200

Model number	IV*a*	V	V*a*	VI	VI*a*	VII
Cooling effect in thousands of B.T.U. per hour:						
From 50° to 34°	238	333	476	675	834	1072
" 28.5° to 23°	199	278	397	556	715	893
Ice production in pounds per hour	830	1100	1650	2200	3300	4400
Ice replacement per day in cwts	300	420	600	840	1100	1400

TABLE IX.

Model number	1	2	3	4	5	6	7	8	9
Cooling effect in thousands of B.T.U. per hour:									
From 50° to 34°	9·5	14·3	19	24	29	48	62	87	103
" 28.5° to 23°	7·9	11·9	16	20	24	40	52	72	87

Model number	10	11	12	13	14	15	16	17	18
Cooling effect in thousands of B.T.U. per hour:									
From 50° to 34°	191	263	382	572	720	880	1080	1320	1600
" 28.5° to 23°	159	220	320	480	600	760	880	1100	1320

Most of the better manufacturers have models corresponding with the numbers given in the tables, so that renewal does not, in general, require any new construction. If the cooling effect required falls between two of the values given in the table, the compressor-cylinder of the next lower model is often bored out to a somewhat greater extent or else its speed is increased. The accessory apparatus (evaporator and condensers) must, of course, correspond with the real yield of the machine and not merely with the data given in the tables, if the machine is to work economically.

CHAPTER III.

THE CONSTRUCTION OF COMPRESSORS.

15. Ammonia Compressors.—Owing to their importance, we shall deal first with the ammonia cooling-machines, for which the most widespread form—that of *Linde*—has become typical. We shall describe only the latter completely, pointing out the varying constructive details of other firms in their proper place.

Horizontal *Linde compressors* are of two types, differing principally in the *guiding of the cross-head.* The best proof of the convenience of their arrangement lies in the fact that, since their first appearance, they have undergone no essential alteration. The guides with a flat, planed cross-head path with a screwed-on guide (after the method introduced by the "Augsburg" Machine Company; see Fig. 17) have an advantage in being clearly visible and readily accessible, but are somewhat more expensive than the cylindrical, bored guides (after the model of Sulzer Bros., Winterthur; see Fig. 18). As also, in the latter, the compressor is fixed to the frame by means of cylindrical fittings, the centring of the compressor with the cross-head centre is rendered easier, but, on the other hand, the whole bed is made somewhat heavier. Consequently the advantages and disadvantages of the two arrangements just about balance. The large double compressors represented in Figs. 17 and 18 differ further in the fact that in the former the cylinder axes are not in line as they are in the latter. This is owing to the attachment to the same crank of two connecting-rods, of which one in Fig. 18 is forked and so encloses the head of the other, this being the only way in which the coincidence of the cylinder axes can be attained.

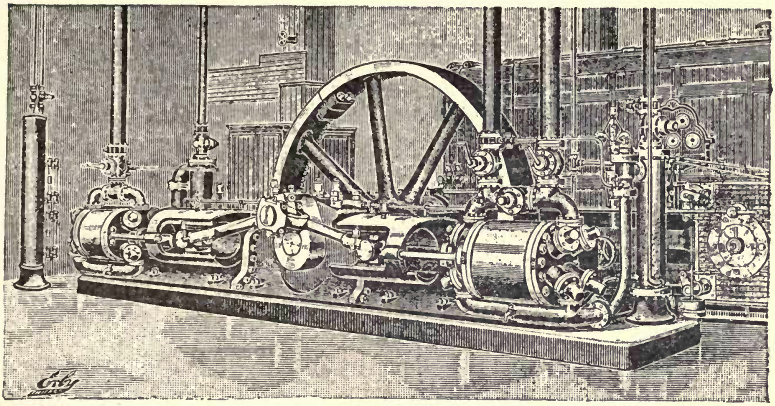


FIG. 17.

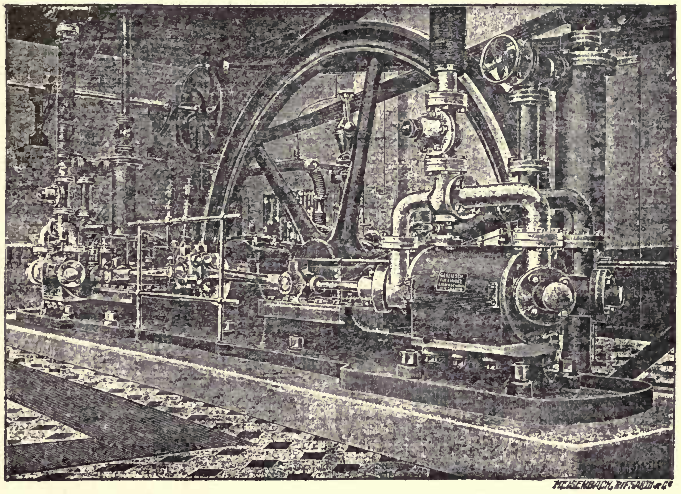


FIG. 18.

Further, one sees in Fig. 18 that between the cylinder and the guide is a massive rod which prevents a tilting of the cylinder fastened to the back side of the frame, and that the bearing for the crank-axle is fixed to the framework of the two cylinders by screws. This arrangement is chosen more especially when only one compressor is required, but a doubling of the plant is expected. Using a bed-plate in one piece for the two cylinders, as represented in Fig. 17, it is necessary to obtain the whole plate in the first place, so that greater expense is incurred. Besides which, the preparation of large plates is accompanied by technical difficulties.

For very large compressors, the *Germania* Company, which formerly preferred the Augsburg arrangement, uses round guides. The compressor represented in Fig. 19 and made by this firm is connected with the very solid guiding arrangement by means of two rods and presents a very substantial appearance. The direct screwing of the guides to the cylinder, as is done in steam-engines, is here, and also in Fig. 18, inadmissible, as the stuffing-box and valves must be accessible.

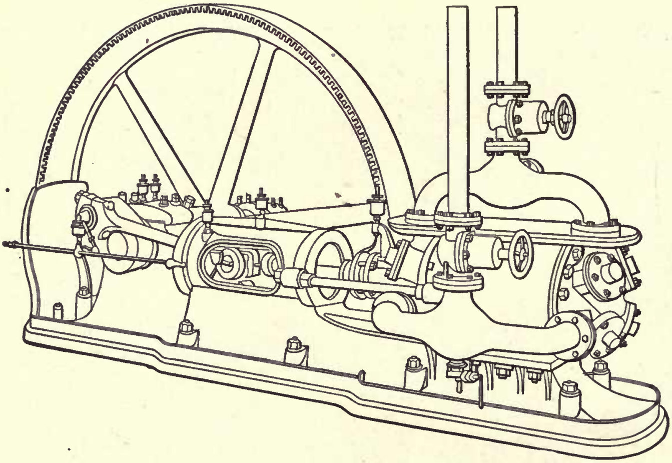


Fig. 19.

In Figs. 20, 21, and 22 are shown the cross-section, longitudinal section, and plan of a medium-sized Linde compressor of Augsburg construction. The simple tube-shaped cylinder, surrounded by a jacket, carries at the two ends covers in the form of spherical caps, on which the valve-boxes are directly cast. The front cover contains also the stuffing-box. The whole rests on planed filling strips of the hollow bed-plate, the foot of which is made in the form of a wide basin for catching both oil dropping down and the water from the frost on the valve-cover which thaws when the machine is at rest. With suction-pipes leading upwards it is advisable to place, immediately

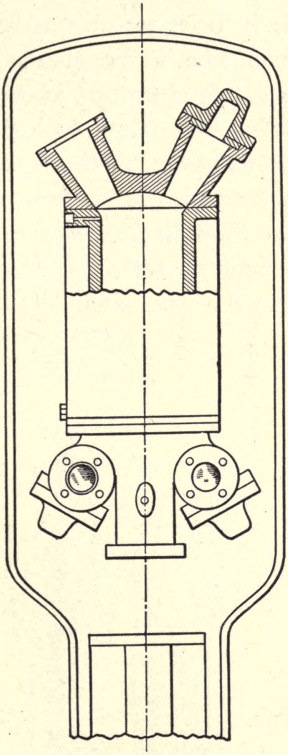


FIG. 20.

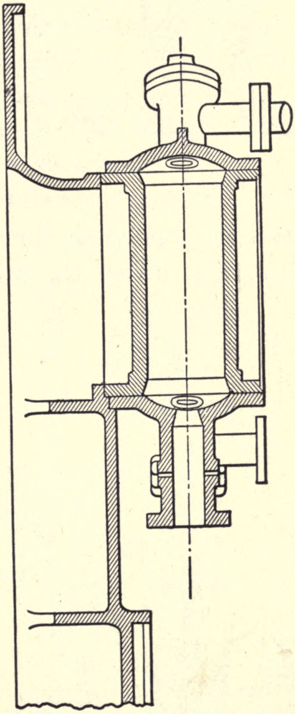


FIG. 21.

over the compressor-cylinder, a drip-pan, to prevent water running down the cylinder-cover and rusting it.

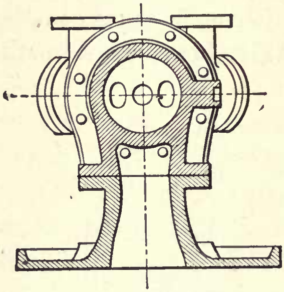


Fig. 22.

The axes of the conically-bored valve-boxes are horizontal and are inclined to the axis of the cylinder at an angle of from 30° to 50°. Only in this way has it been possible to give the valves a sufficient sectional area without increasing the clearance in the cover and at the same time to leave room for the stuffing-box in the front cover. The steel *valves* themselves are flat plates (see Figs. 23 and 24), the spindles of which carry only one shoulder each to take up the pressure of the spring. With the suction-valve this shoulder is placed at the outer end of the spindle, but with the delivery-valve where it takes part also in the guiding it is fixed at about the middle of the spindle. Since it is turned in one piece with the spindle, the guides for the latter must be made in two pieces

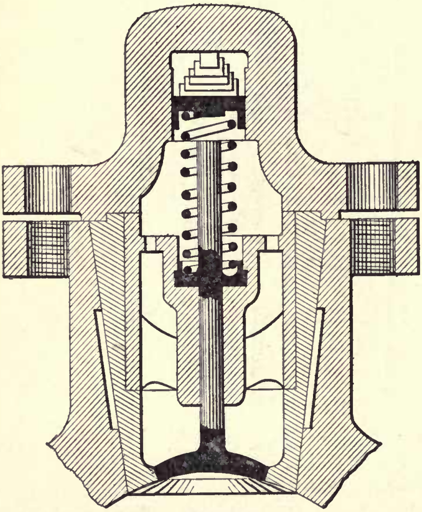


Fig. 23.

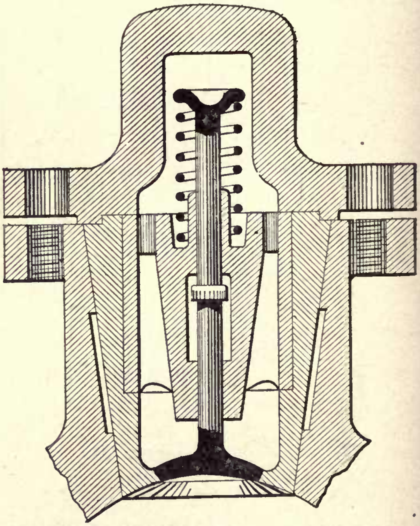


Fig. 24.

to allow of its insertion. They are strengthened by ribs and abut on the real valve-case, which, like the box, is conically turned and is pressed into the latter by the cover. The valve-seatings are of fine-grained cast iron. The plates are spherical towards the interior of the cylinder, so that, without giving rise to useless space, they completely conform to the spherical form of the cylinder-cover. In order to diminish the danger of the suction-valve falling into the cylinder, its spindle has, in recent times, often been provided with a narrow journal in the middle, and for this the guide must of course possess an enlargement. Further, for catching up the pressure-valve, a plate in the cover supported by a conical spring is very suitable.

When the delivery-valve closes, the spring-plate, which must fit well in the guide-box, serves as an air-buffer. Its action can

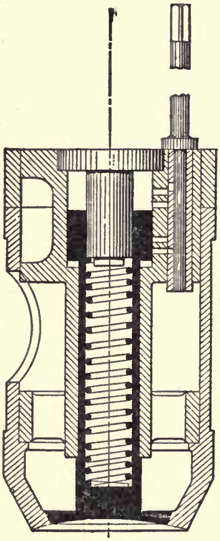


Fig. 25.

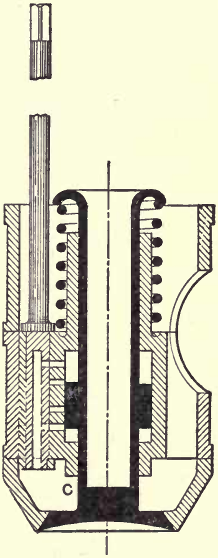


Fig. 26.

—as shown in the arrangements of Seyboth of Munich and of the Atlas Company of Copenhagen, represented in Figs. 25 and 26—be regulated from outside by constricting or widening the connecting channel of the buffer-box by means of sliders. The adjustment of the sliders is best done from the indicator diagrams, from which the proper valve-aperture is ascertained, while at the same time the noise made on setting the plate is observed. Incompetent adjustment of these sliders should be prevented as far as possible, so that the closing of the valves may not become slipping and considerably delayed.

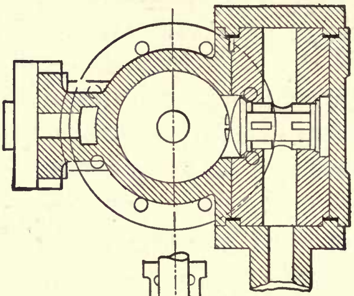


FIG. 28.

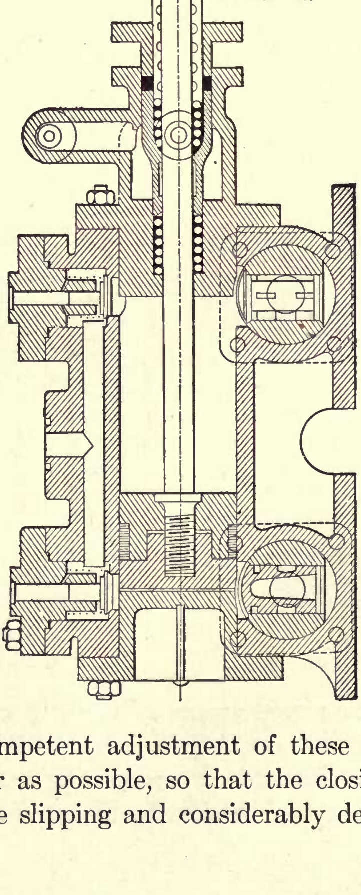


FIG. 27.

The use of valves without separate chambers, as in Figs. 27 and 28, does not seem very convenient, as it must be very difficult to centre the seating for the plate with the cover in which the spindle of the pressure-valve moves. With the suction-valves made by *Kilbourn* this error is avoided. These are placed in special chambers, which, like the Corliss valves, are set in the cylinder-foot, so that separation of the branched pipes is unnecessary. A disadvantage of this arrangement is a considerable increase of the clearance.

Having regard to the exact play of the valves and the resistance to motion in the pipes, the mean *piston velocity* is not allowed to exceed 3 feet per second, from which the ratio of the diameter of the cylinder to the stroke is given as 0·6 to 0·75, according to the size of machine. The free *total section* of the suction-valves and the suction tubing should then be $\frac{1}{10}$ $\frac{1}{12}$, and that of the delivery-valves and pressure tubing as far as possible not less than $\frac{1}{15}$, of the mean active piston surface of the compressor, the corresponding steam velocities being 32–48 ft. per second. During the short time when the valves cross, it is impossible to prevent these velocities from rising to twice or thrice these values, it being found that the valves, especially under the influence of strong springs, do not always attain a stroke equal to one-quarter of their free section, as is required by theory. On this account it is convenient to choose the springs as weak as is possible without endangering the *rapid closing* of the valves.

The hollowing-out of the cover corresponds with the shape of the *piston* (Fig. 29), the latter consisting of two hollow castings, which are pressed together by a sunk screw on the end of the piston-rod and so hold fast the piston-rings and the spring underneath them.

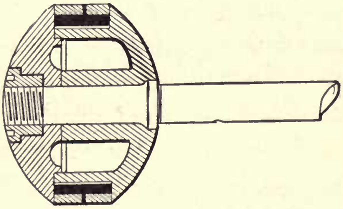


Fig. 29.

As can be seen from Figs. 17 and 19, the tubes from the suction- and delivery-valves are joined in pairs, forming a suction-pipe and a delivery-pipe, these being each provided with a shut-off disc or cock. These latter, like all the above-mentioned, must consist of iron, since copper, bronze, and brass are all attacked by ammonia; their form can be seen from the accompanying drawings, of which Fig. 30 represents a slide from a *Linde* machine and Fig. 31 a cock of the *Kilbourn* construction. In the common delivery-pipe, yet in front of the shut-off cock, is a small cock with a tube (visible in Fig. 18, where the delivery-pipe runs below the compressor), through which the compressor can be quickly emptied—the shut-off cocks in the suction- and delivery-pipes being closed—before repairing or overhauling it. When doing this, it is simplest to lead the ammonia through a rubber tube into water, by which it is rapidly absorbed.

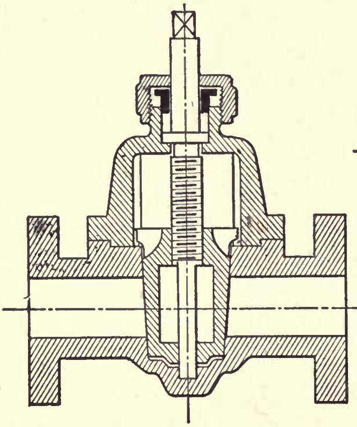


FIG. 30.

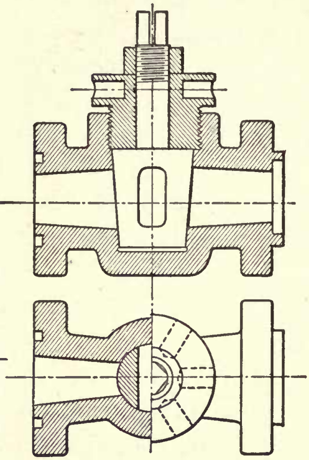


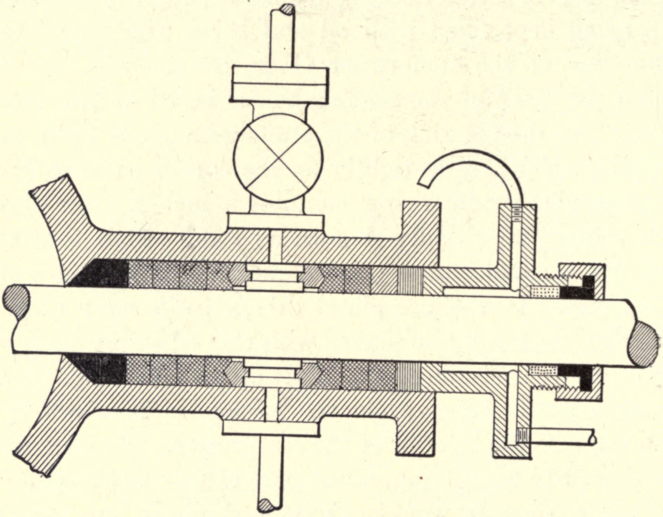
FIG. 31.

One of the most important components of the compressor is the *stuffing-box,* the purpose of which is to prevent not only the escape of ammonia, but also the access of atmospheric air or water (liquid or vapour) to the interior of the machine. The cycle of the ammonia being a closed one, the suitable construction and maintenance of the stuffing-box determine to a very large extent the safe working of the machine. *Linde* was the first to arrive at a perfectly satisfactory solution by the use of the so-called lantern gland. This consists of a double ring joined by ribs and not actually touching the piston-rod; this ring lies between rings of cotton meeting in oblique ends, the intermediate space being kept filled with oil and being also joined to the suction-pipe of the compressor (Figs. 32 and 33), for which purpose the neck of the stuffing-box is bored at two opposite points. The double ring of the stuffing-box is of white metal; the gland, which presses tightly on the cotton packing through an intermediate rubber ring, contains a hollow space through which compressor-oil (also known as Baku oil) is continually forced by a small pump (placed directly under the stuffing-box and fixed to the bed-plate) driven by a strap (Fig. 33). The cotton plaits are also soaked with this oil before their introduction into the stuffing-box. Part of the oil runs down again through a small tube, while the remainder is drawn by the piston-rod into the interior of the compressor. By contrivances to be described later, the greatest part of this latter oil is separated again, since it would otherwise diminish the passage of heat through the tubes of the condenser and evaporator.

The oil vessel (Fig. 33) connected with the pump should further be carefully protected against the entry of water, especially that which forms on the cold outer walls on the suction side and drops down when the machine is not working, since this would go with all its impurities through the oil-pump into the machine and have a serious effect on its action.

Linde's stuffing-box was later altered by *Fixary* (see Figs. 34 and 35), who considerably widened the oil-chamber and provided the stuffing-box flange with an annular hollow space, separated air-tight from the exterior, in which any ammonia coming from the condenser through *a* is allowed to evaporate by means of a regulating-cock. The cooling action produced in this way causes the oil in the cotton packing to freeze and so gives a tight fit at the outside. Since, however, it is found in practice that, with care, the stuffing-box becomes tight without this freezing process, this transference of ammonia is usually omitted, and the working of *Fixary's* stuffing-box differs from that of *Linde* only by the immediate introduction of the oil from *c* and *d* into the hollow space at the lantern, which is also connected by *e* with the suction-pipe.

FIG. 32.



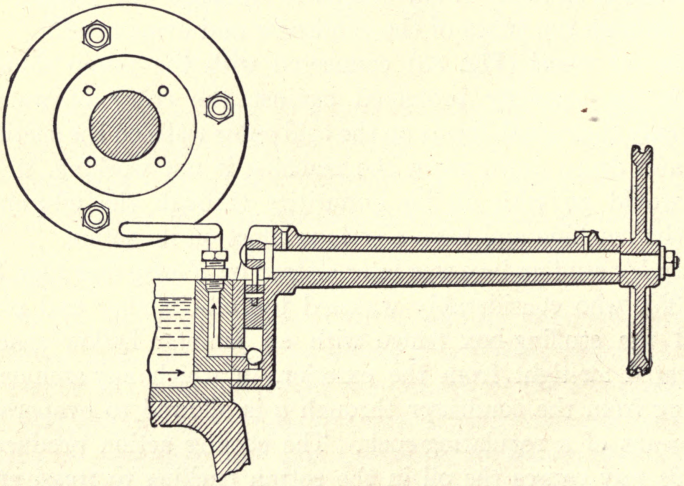


FIG. 33.

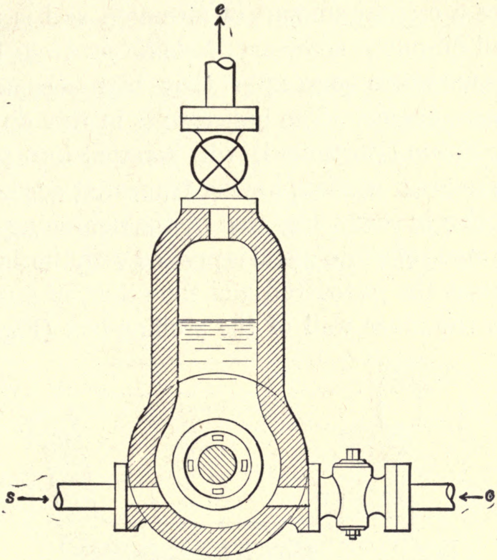


FIG. 34.

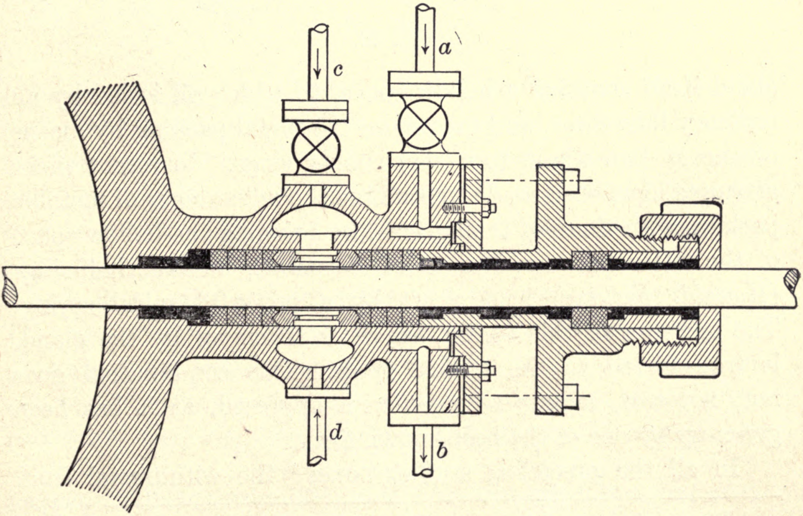


FIG. 35.

Since this continual transport of oil through the stuffing-box requires a certain amount of attention and is complicated by the small oil-pump necessary, *metallic packings* began to be used for compressors soon after they had become successful with the steam-engine. The best results in this direction were obtained by *Friese* (Dortmund), who screwed into the box two spirals of triangular section, having truncated edges and fitting into one another in such a way that by the tightening of the gland one of them made of white metal is pressed with the flat face of the triangle against the piston-rod and the other, of steel, with the face against the inner wall of the stuffing-box (Fig. 36). The gland itself contains a hollow space, to which oil is led by an ordinary lubricator, and presses on the metal packing by means of one or two whole (not slit) rubber rings. In recent times attempts have been made to combine the advantages of metallic packing with those of the lantern, by bringing the hollow space of the latter between two series of conical, metallic half-rings (*Howaldt packing*) held apart by a powerful spiral spring. The pressure of this spring, which is taken up by the gland, is transmitted to the conical rings. This combination does not, however, appear to meet with approval, as it has been given up by one of the best-known firms.

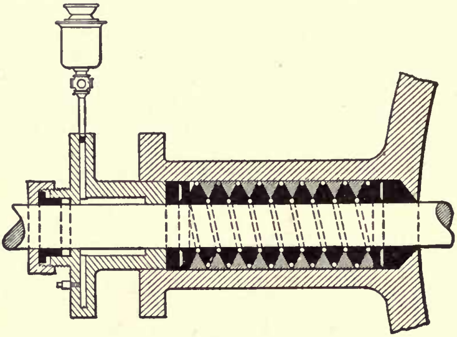


FIG. 36.

In all the preceding stuffing-boxes[1] the withdrawal from the gland of the oil in which all the tightening rings are soaked before use is effected by a thin felt packing held fast by a ring and a nut.

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[1] For certain other forms see the paper by C. Schmitz, Zeitschrift für die

The permanent tightness of all stuffing-boxes requires that the temperature shall not rise too high during compression. If this cannot be avoided, owing to the high temperature of cooling-water (in the tropics), *the tightening arrangements must be distributed between two cylinders*, of which, according to the method adapted by the Linde Company (Fig. 37), the low-pressure compressor *A* contains the stuffing-box, while the high-pressure compressor *B* is placed at the back. The intermediate space *D* is joined with the suction-tube of the low-pressure cylinder, while the chamber *C* is arranged over the two cylinders. The main construction of the machine is not appreciably altered in this way.

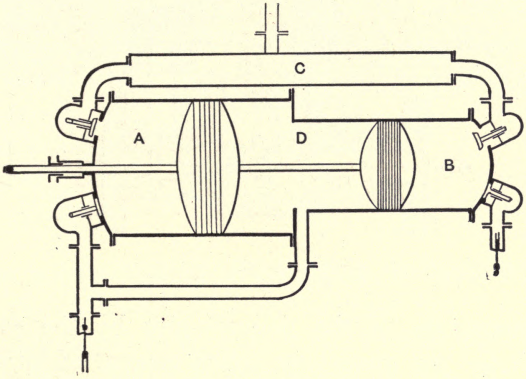

FIG. 37.

16. Regulation and Oil-separation in Ammonia Machines.—The *regulating-valve*, which effects the throttling during the passage of the liquid cooling-agent from the condenser to the evaporator, consists in most ammonia machines of a normal valve-cone (see Fig. 38), which is rotatable on its spindle and

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gesammte Kälte-Industrie, 1895, parts 9-12, from which also some of the above figures are taken.

carries another longer, very narrow cone. The latter does the real regulation, since it determines the area of passage according to the height of stroke. The adjustment of the valve is effected by means of a hand-wheel, while the operator observes either the manometers of the condenser and evaporator or the temperature of the delivery-pipe by the compressor, which, with the *wet compressor action* general in former times, can be conveniently done by touching with the hand. With the *superheated compressor action* favoured of late years[1] —which has been found not only harmless for the machines but also more advantageous for their economical working—this simple control is lacking and the only thing to do is to observe the manometer, if there is no thermometer inserted in the delivery-pipe behind the compressor.

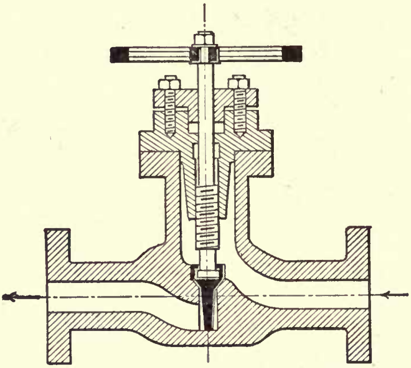

Fig. 38.

In order to render the stable condition of the machine as far as possible independent of the engineer, an attempt was first made to make the position of the regulating-valve visible from outside by means of a scale on the rim of the hand-wheel. *Linde* proposed another method: he fashioned the regulating-valve into a rotating cock whose plug (see Fig. 39) has a rectangular channel permanently connected with a space *R*, the volume of which can be altered by the hand-wheel *H* and piston *B*. In the course of the rotation, which is derived from the compressor and transmitted by the pulley *S* to the spindle *A*, the space *R* is first filled from the tube *K* coming from the condenser and then gives up its contents to the evaporator-tube *V*. The regulation is here limited to adjusting the hand-wheel *H* once and for all. Although this arrangement found favour everywhere, *Linde* sought to improve it still further and arrived at the regulating-valve which is represented in Fig. 40[1] and which becomes at the same time a *distributor* of the ammonia to the separate worms of the evaporator. The liquid ammonia passes into the hollow space of the foot and there comes to rest, so that oil carried along with it may settle on the bottom and be drawn off from time to time. The plug of the cock is here rotatable vertically and, by means of an aperture in the jacket, connects the cavity once in every revolution with each of the evaporator-worms having their openings in the chamber. In consequence of the regular distribution in the tubes, their heating surface is better made use of and, so long as the apparatus is in good condition, i.e., the plug is tight in the box, a considerable increase (up to 20 per cent) in efficiency is obtained.

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[1] This takes place if the compressor draws in quite dry saturated vapour, as was assumed in our examples in Chapter II.

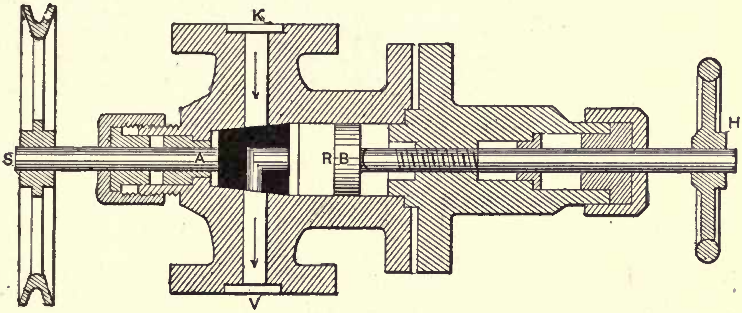

FIG. 39.

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[1] Given with the corresponding figure in Prof. *Linde's* article on Cooling Machines in Lueger's "Lexikon der gesammten Technik."

Of very considerable importance for the working of ammonia machines is the greatest possible *prevention of the access of compressor-oil* to the tubes of the condenser and evaporator, which would otherwise become greatly injured in their action. The separation of oil from the ammonia is rendered somewhat easier by superheated compressor action, because the ammonia,

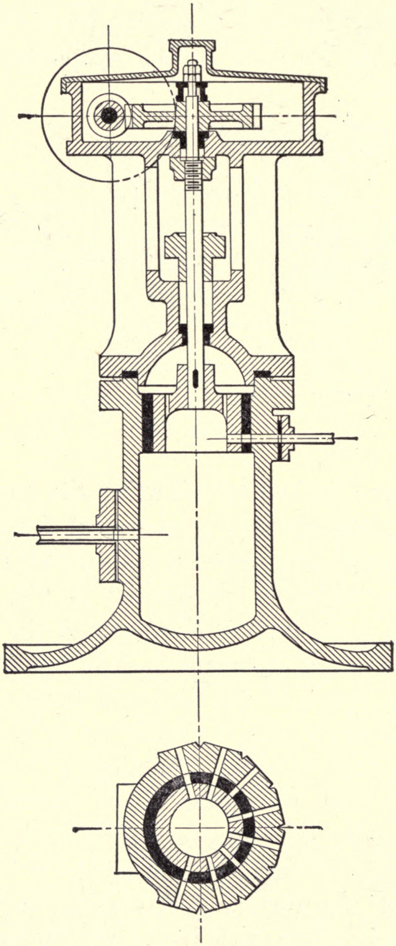

FIG. 40.

leaving the compressor at a higher temperature, is only absorbed or held back by the oil to a small extent. The separation is accomplished in a *cylindrical vessel* (Figs. 41 and 42), which is placed close to the compressor by *Linde* and most other makers, and in this empties after passing a return-valve.

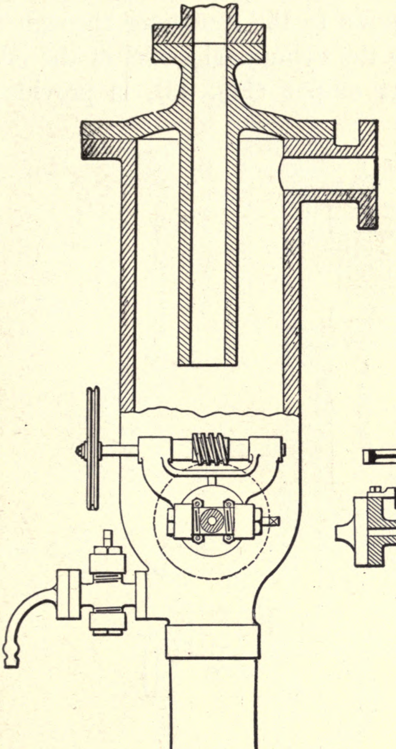
FIG. 41.

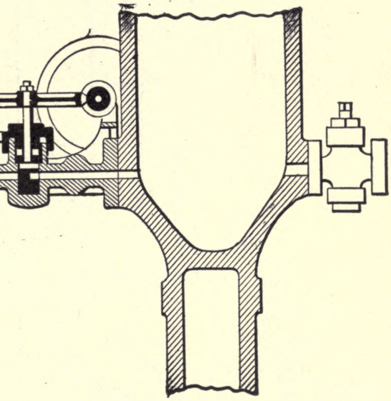
FIG. 42.

On the bottom of the vessel the oil collects and is carried to an oil-collector by a tube in which a constantly rotating cock allows of the passage of a certain quantity of oil each revolution; the oil-freed ammonia flows to the condenser through a tube introduced centrally into the cylindrical vessel of the oil-separator. The oil-collector, or oil-pot (Fig. 43), is provided with an open jacket, which can be filled with warm water (condenser waste, for example) to expel the ammonia from the oil in the interior. The ammonia here evolved passes into the evaporator, or, better, into the suction-pipe of the compressor.

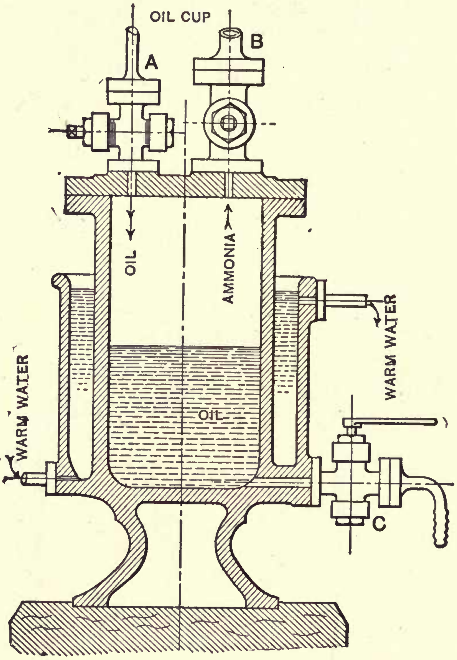

Fig. 43.

The Machine Company, formerly *Klett* and Co., of Nürnberg, have combined the oil-separator and collector in a very neat way by making the foot of the former into the oil-pot (see Fig. 44), which they provide with a double bottom for the passage of hot water to expel the ammonia. The transference of the oil from the separator to the collector takes place through a hand-regulated cock the action of which is observed by means of glasses (see Fig. 45).

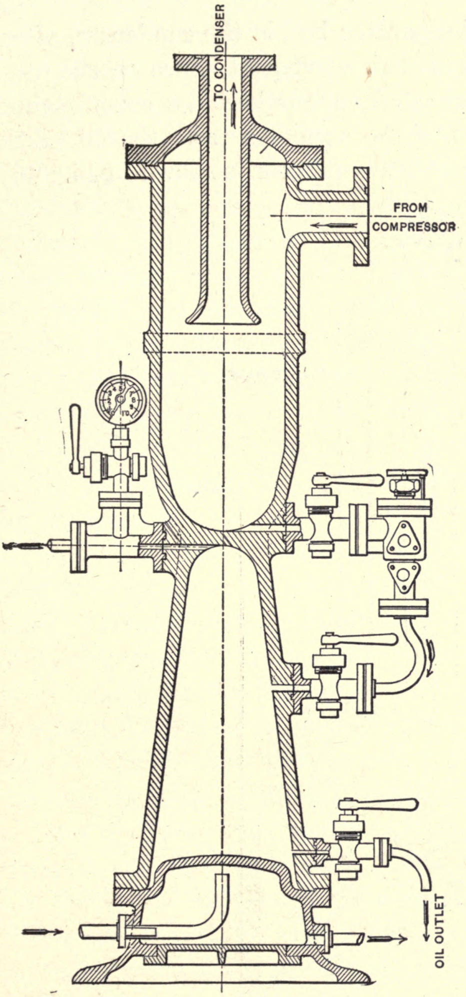

Fig. 44.

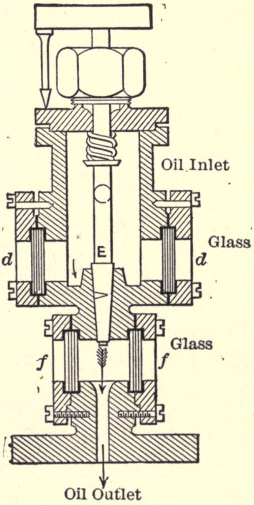

Fig. 45.

The above-described separators are placed immediately behind the compressor in the delivery-pipe so as to protect this from the oil.

The placing of the oil-separator beside the condenser, after the method of *Fixary*, seems less convenient, since on the way thereto the initially superheated vapour becomes considerably cooled and the separation of the ammonia from the oil takes place partially in the pipe. The separator is shown in Fig. 46; it consists of a cast-iron body, widened at the bottom, with a central entry and side exit for the ammonia, and has, inside, a cylindrical sheet-iron vessel, funnel-shaped and provided with

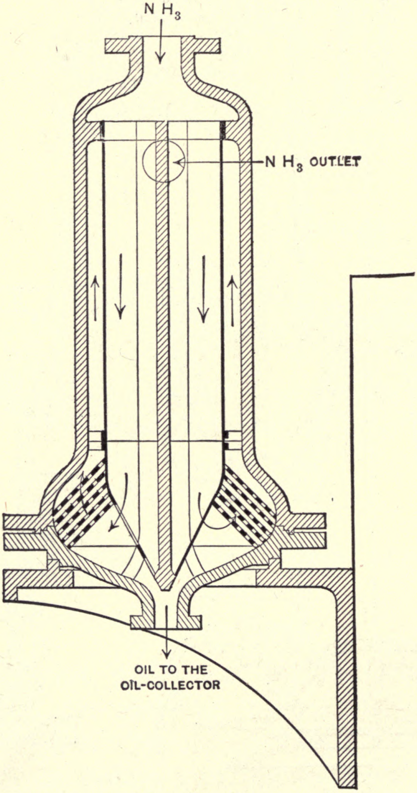

FIG. 46.

apertures at the lower part. In the wide portion and beside the funnel are sieves, through which the ammonia has to pass and to which it gives up its oil, which leaves the separator at the lower end and proceeds to the oil-collector. The latter is a cast-iron vessel, furnished with an internal heating coil for expelling the ammonia. This separator has a more involved construction than the two previously described.

Owing to its relatively large volume the oil-separator exerts, moreover, a favourable influence on the compressor as an air-chamber. They take up the rapidly and intermittently expelled contents of the compressor without appreciable increase of pressure and pass them on regularly to the condenser, thus preventing any considerable augmentation of the excess pressure in the pressure-valves of the compressor. Of course, this desirable action only takes place undiminished in amount if the oil-separator is placed in the immediate neighbourhood of the compressor-cylinder and if it is connected with the latter by a relatively wide tube.

In order to prevent solid bodies, e.g., iron scale, from being carried along from the pipes into the compressor, the suction-pipes of all machines are fitted near the compressor with a *vessel containing a sieve.* This must be looked to and cleaned from time to time, so that the suction resistance may not be rendered too high by partial stoppage or the sieve itself torn. This disadvantage is entirely removed by means of a slit in the sieve opposite the exit. Figs. 47 and 48 represent such a dirt-catcher of the cylindrical form chosen by the *Linde* Company; the ammonia enters at *a* and passes out at *b* to the compressor. At *c* is the end of another tube, which brings back any ammonia which has leaked from the stuffing-box or separated in the oil-collector, while at *d* the tube of the suction-manometer is generally affixed.

Finally, in the delivery-pipe *Linde* places an ordinary *recoil-valve,* which closes in case of a sudden fracture (e.g., of the cylinder-cover) and prevents the contents of the condenser from flowing back.

It is seen from the foregoing that, strictly speaking, three different cycles are performed, these being that of the *circulating ammonia* indicated in Fig. 49 by a thick line ——; that of the *oil*, which is indicated by — - — — — — - —, and which, coming partly from the stuffing-box and partly from the oil-separator, collects in the oil-pot *c*; the *expelled ammonia* passes from the latter by a path indicated by a faint line, ———, to

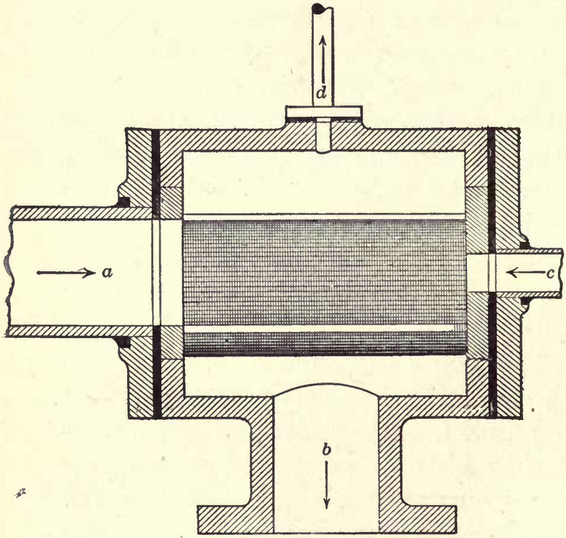

FIG. 47.

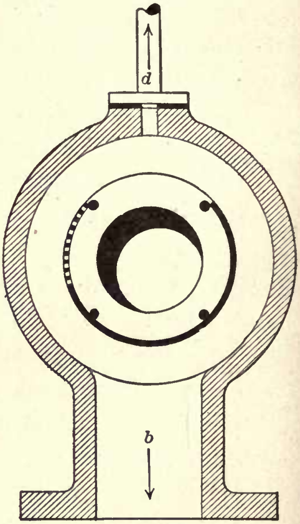

FIG. 48.

the dirt-catcher of the suction-pipe. The figure shows also the path taken by the ammonia from a cylinder for filling the machine; this path joins the tube (indicated by — — —.—) connecting the condenser and evaporator, after the regulating-valve. It is assumed that waste cooling-water from the condenser is used for heating the oil-collector, the two pieces of apparatus being joined by a line - — — — - — — —. The manometer-tubes are indicated by — - - - - - - —.

In America, where superheating of the compressor is not infrequently carried on to a high degree, cooling of the compressors is often combined with the introduction of oil, the latter being sprayed into the cylinder and, after separation, cooled down by a special oil-cooler (*De la Vergne* Process).

The oil separating from the machine is nearly always mixed with fresh and used again. This should, however, only be done after filtration, and special care should be taken that no water is carried along with it into the machine.[1] Admixture with water readily takes place, however, in the dropping-tray of the stuffing-box (Fig. 17), even when, with cold working, this box

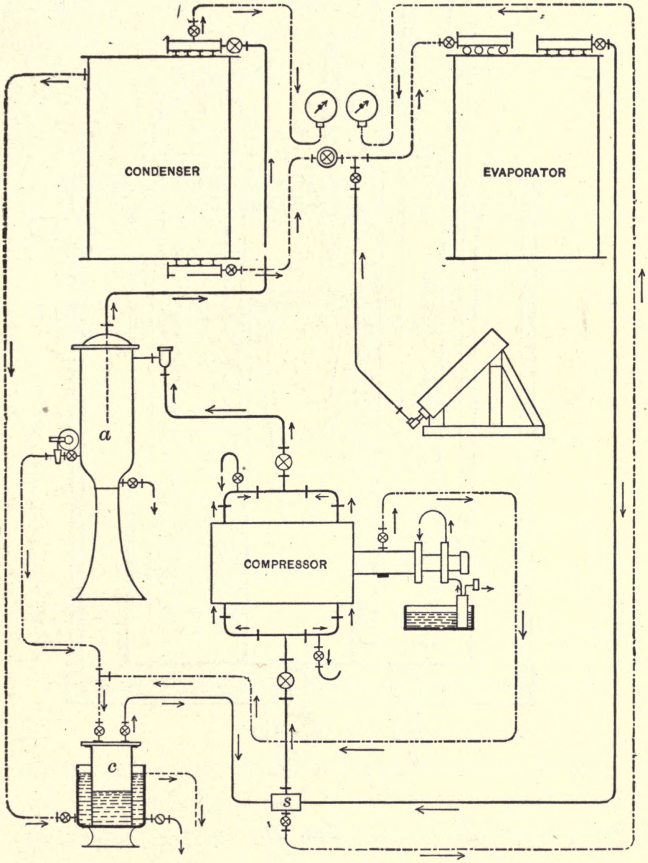

FIG. 49.

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[1] On the influence of impurities of ammonia, see § 7, above.

is lagged outside. The separation of the water from the oil is most simply effected by strongly cooling the mixture, whereby also the resinous constituents of the oil solidify and can then be removed by filtration. A neat combination of such an apparatus with the cooling arrangement (*B. Weiser*, Basle) is shown in Fig. 50. In an iron-plate vessel, *a*, well insulated with wood, are evaporator spirals, *k h i n*, through which the cooling-agent itself is passed according to requirements. The oil to be purified is introduced into this vessel through *e* and rises until,

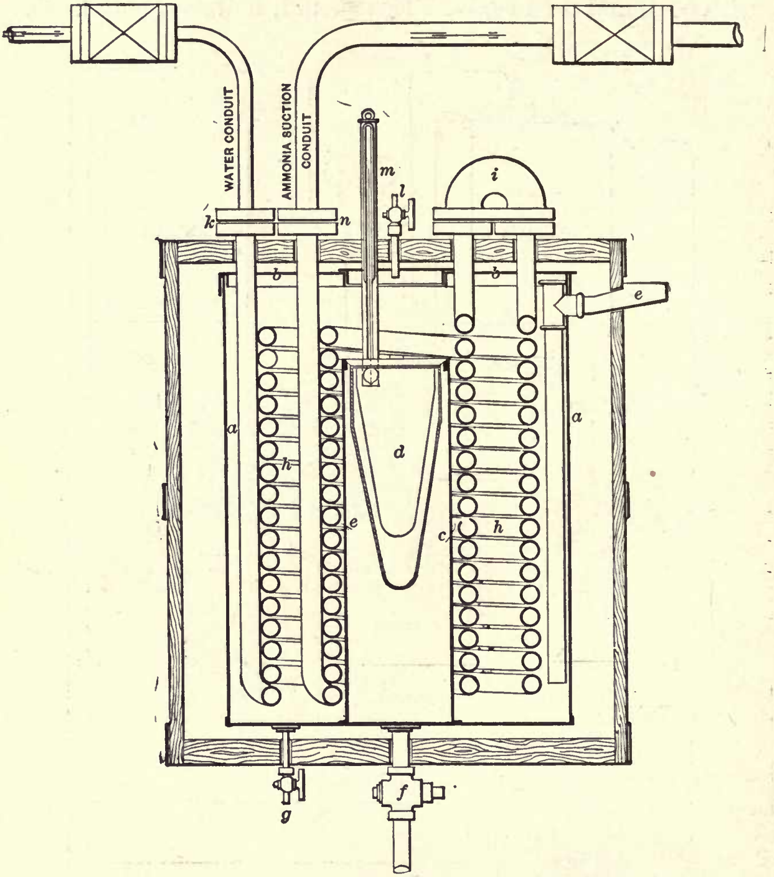
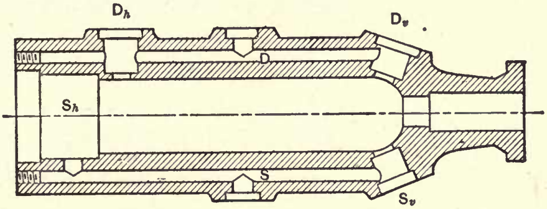


FIG. 50.

cooled to a low temperature, it reaches the inner space provided with a sieve and filtering material *d*; from this space it is run off clean through the cock *f*. The apparatus can be emptied by the cock *g*, evacuated through *l*, while a thermometer *m* renders control easier.

17. Carbon Dioxide Compressors.—The fundamental requirements in the construction of these compressors are: Tightness and capability of resisting very high pressures. The latter of these necessitates the cylinder being made of tough, bubble-free material (very fine-grained cast-iron or cast-steel) and of small diameter, so that the strength of the walls need not be inconveniently great, as this would injure the homogeneity of the casting. Tightness towards gases requires, besides faultless material, as small a number as possible of tubulures and stuffing-boxes. Otherwise than in ammonia machines, where each separate valve is connected outside with the corresponding tube, the number of tubulures is here generally diminished by the

FIG. 51.



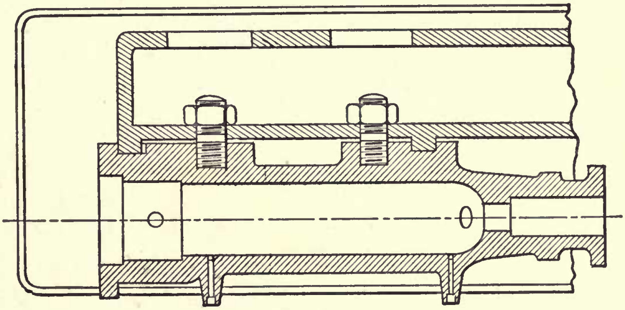


FIG. 52.

delivery-valves—which are almost always arranged at the top—having a common delivery channel cast into the cylinder (see Figs. 51 to 56).

Of the suction-valves one is almost always joined to the back cover, while the front side contains either one pointing obliquely downwards or two oblique side suction-valves, for the accommodation of which the front part of the bottom of the cylinder is given a spherical shape. The first arrangement (Figs. 51–53, made by *Escher, Wyss & Co.*, of Zürich) is generally found with cylinders which are connected to the side of the frame by means of turned fitting-strips, so that the under side of the cylinder is left free for a central tube and for a front suction-valve directed downwards. A common suction-pipe is, of course, used here.

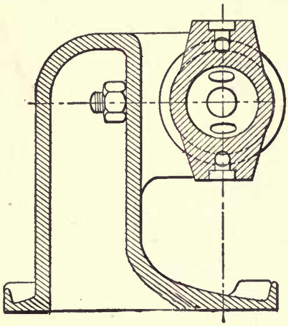


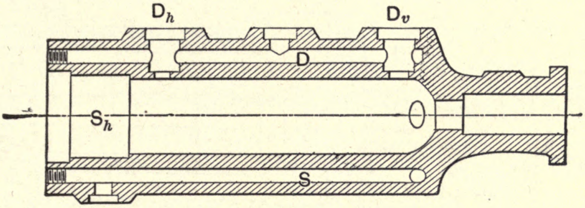
Fig. 53.

In the second arrangement, utilised by the firm of *L. A. Riedinger*, of Augsburg, it is no longer practicable to place the front suction-valve below, and it becomes necessary to make use of two side ones (Figs. 54 to 56). The suction-channel for the valves is likewise cast on the under side of the cylinder and joined near the back end with the tubulure. It would also be easy to fix the suction-valve of the rear cover directly to the latter; but it is more usual to provide it with a special side aperture, as otherwise it would have a harmful effect on the action of the front valve, on account of its unsymmetrical disposition as regards the suction-channel.

For the more recent cylinder construction of *Halle's* Union (formerly *Vaas & Littmann*), shown in Fig. 57, the suitable and exact treatment of the interior of the cylinder is of importance. The four valve openings are bored in exactly the same way, so that changing is rendered very easy. The front cover contains the stuffing-box, but no valve.

The sketches, which will be understood without further description, represent, for these two principal arrangements,[1]

FIG. 54.



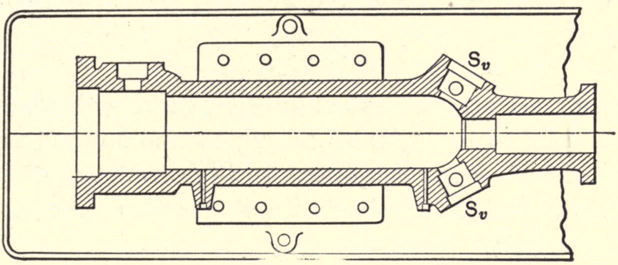


FIG. 55.

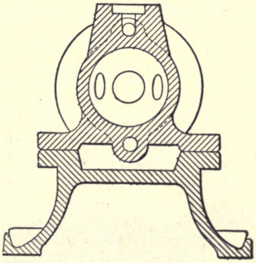


FIG. 56.

the casting for the cylinder and the method of fixing it to the stand. *S* and *D* are the suction- and pressure-channels respectively, S_v and D_v the suction- and pressure-valves in front, and S_h and D_h those behind. The cylinder is generally provided with a covering of iron-plate, above which only the indicator openings, the tubulures, and the flanges for the valve-covers project.

Owing to the high pressures of carbon dioxide, the *stuffing-boxes* at first presented great difficulties, more especially as the piston-rods are stout compared with the diameter of the cylinder.

---

[1] These arrangements correspond perfectly with those previously given for ammonia machines.

Just as with ammonia machines, in this case also it has been found that the most perfect tightness is obtained by the introduction of a lantern (Fig. 57) filled with lubricating material (mostly glycerine); the carbon dioxide reaching this lantern is then returned to the cylinder by joining the lantern with the suction-channel of the compressor. In this manner the pressure prevailing in the lantern is always the suction pressure (350–430 lbs. per square inch).

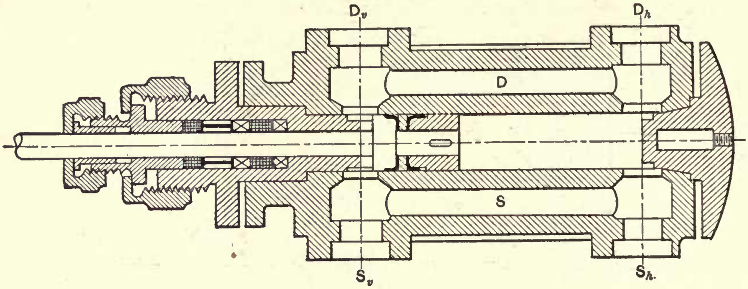


Fig. 57.

For packing, rubber rings with numerous radial insertions of cotton have given good results, provided that the lubrication is effected with compressor-oil. An excellent packing introduced by Messrs. *Riedinger*, of Augsburg, for use with pure glycerine lubrication and largely employed during recent years, consists of *leather collars* with *rubber rings behind them.* Fig. 58 represents a complete stuffing-box with this arrangement. The

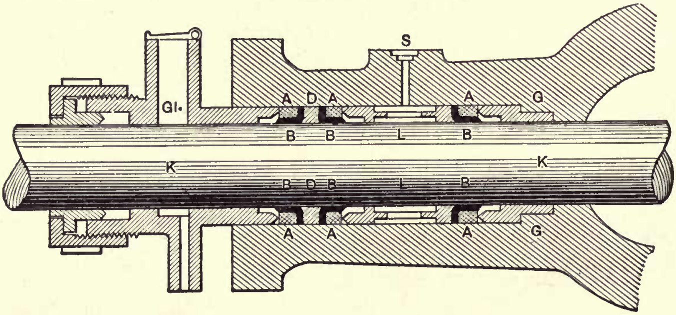


Fig. 58.

leather collars are naturally not elastic, but are kept flexible by the glycerine carried along by the piston-rod *K* from the glycerine chamber *Gl* and are pressed against the rod by the rubber rings *A* behind them. The metal parts of the stuffing-box, that is, the base ring *G*, the lantern *L*, and the pressure-ring *D*, consist of bronze and the rod itself of steel. In fitting up the rings, and also the front box, with the glycerine chamber, it must be made certain that they slide quite easily over the polished rod. The cavity of the lantern *L* is finally to be connected with the suction-channel by means of the tubulure *S*.

In order that the glycerine may not be carried to the outside with the piston-rod, a simple stuffing-box, with a thread for tightening it, is placed in front of the chamber *Gl*; the packing of this box (ordinary felt) need not be capable of standing any appreciable pressure.

In the practical adjustment of this arrangement, it is only necessary to take care that the box which contains the glycerine chamber *Gl* is tightened (by three screws) just sufficiently and quite uniformly, so as to avoid all corners. Excessive tightness must be avoided, because the rubber in becoming saturated with carbon dioxide swells considerably and so presses strongly on the leather. It can be seen whether the stuffing-box is in good order by opening the cover of the glycerine chamber; small bubbles should here be visible from time to time, but a permanent vigorous foam should not make its appearance.

All loss of carbon dioxide through the stuffing-box is prevented in a very neat way by Messrs. J. and E. *Hall* of Dartford (England), whose carbon dioxide machines are mainly employed on ships. They make use of a small cylinder with a piston (Fig. 59), whose rod passes out on one side through the cover into the air. This side contains glycerine, which can be replaced from time to time and is connected with the stuffing-box lantern of the compressor by means of a tube *C* provided with a throttle-valve, the pressure behind the piston in *B* being made equal to that of the condenser by the tube *D*. The glycerine—which only passes into the stuffing-box in small quantities—is in equilibrium with this condenser pressure and so supports a pressure which is higher in proportion as the front area of the piston (diminished by the rod) is less than the area of the back surface. When this subsidiary cylinder has a diameter of 1½ in., the area of the piston is 1·77 sq. in., and the total pressure on its rear face, under a condenser pressure of 70 atmospheres, 1·77×70×14·7=1821 lbs. On the protruding spindle, 0·5 in. thick and 0·196 sq. in. sectional area, the atmosphere exerts a pressure of 2·88 lbs.; the total pressure on the enclosed glycerine is hence 1821−2·9=1818 lbs., which, acting on the front face of the piston, which has an area of 1·77−0·20=1·57 in., corresponds with a pressure of 1818 : 1·57 =1141 lbs.=77 atmospheres. The lantern is hence under a higher pressure than is ever reached in the machine, so that glycerine finds its way through the latter but carbon dioxide can never be evolved. So far as we know, this ingenious arrangement has not yet been made use of in Germany.

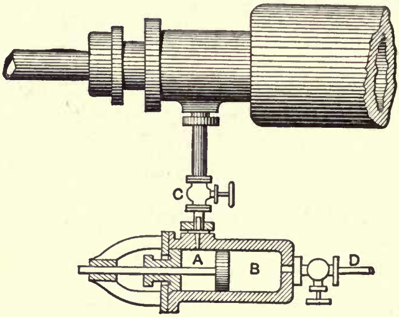


FIG. 59.

We may also remark that Messrs. *Hall* & Co. make their compressors out of massive steel blocks, which scarcely appears justifiable in view of the very great perfection of the technique of casting, which in Germany and Switzerland is able to satisfy the most stringent requirements.

Another organ of importance for the regular working of the machine is the *piston*, which in double-acting compressors must be rendered tight on both sides. Metal rings have not been found to give good results for this purpose, and it is most frequently the custom to use pistons with double leather collars arranged between bronze rings to point in opposite directions and pressed tightly together by a nut of soft wrought-iron secured by a pin. A simpler method is to press the back pressure-ring by a wedge, as in Fig. 57. The leather packings tighten better the greater the difference between the pressures on the two sides, since the carbon dioxide has an opportunity to penetrate behind the collars. In the construction of the piston, it is convenient to make the bronze parts to fit the cylinder closely and the harder iron parts so that they have about $\frac{1}{64}$ inch play and do not therefore touch the walls when working.

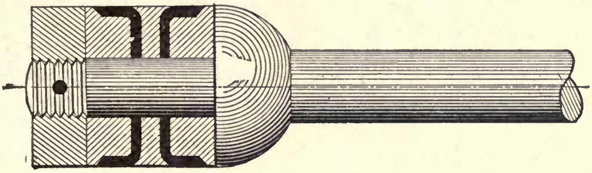


Fig. 60.

The collars consist of neat's leather, 0·15 to 0·2 inch thick for the piston and 0·1 inch for the stuffing-box; they are prepared by hand in suitable *collar-presses* (see Figs. 61 and 62), which are made up of a base piece with screw and nut and a press-ring.

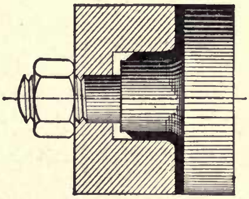


Fig. 61.

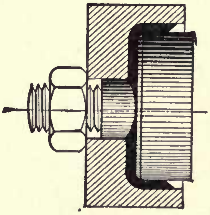


Fig. 62.

When such presses are supplied with each machine, the engineer is in a position to renew the leather collars at any time, but the rubber rings must be obtained accurately cut from the rubber factory.

The life of the leather collars is naturally shorter than that of metal rings; with proper treatment a set of such packings should last through a working season (March to the end of October), but there are examples known where they have lasted longer. For this it is essential to avoid any great superheating in the compressor, since at temperatures of 160–175° F. leather becomes brittle and tears. To a capable engineer this offers, however, no great difficulty.

For the *suction-* and *pressure-organs* of carbon dioxide or ammonia compressors the best results have been obtained with simple plate-valves. The main point to be observed is that the valve-seating should always be joined to the box, which either contains the guide (this being as long as possible) for the valve-rod or has this guide fitted exactly centrally in it by means of long cylindrical or conical surfaces. Where the box and guide are separate, they should never be screwed together, but only placed loosely in contact and then pressed tightly together by the cover. The valve-plates are pressed on to their seatings by springs which should not be too strong; when the rod is vertical or nearly so these springs might be omitted, but it has been found that even in this case they give a certain security to the valve action. If the valve-spindles are horizontal, the springs must be stronger in order to overcome the friction in the guides.

The valves shown in Figs. 63–66 are constructed on these principles. The suction-valve of Fig. 64 and the pressure-valve shown in Fig. 63 are characterized by the fact that the plate T for taking up the spring pressure is made in one piece with the rod and the valve-cone. A consequence of this is that the guides for the rod require adjusting *from time to time.* With very accurate workmanship this construction has given good results, but care must be taken that, when the valve has a horizontal or inclined position, the dividing plane of the two halves of the guide which passes through the axis of the rod does not become vertical, otherwise there gradually forms on the spindle a ridge which can easily cause the valve to clutch.

Further, the opening *C* in the cover closed by a bronze nut, which is frequently found in this type of valve, only serves to render the valve-springs controllable without removing the whole cover and may hence be dispensed with. The casing *B* with the conical (45° to 60°) valve-seating inclined to the axis is bored by the apertures *O* for the entry and exit of the cooling-agent and is generally situated in a cavity of the cylinder into which the suction- or pressure-channel opens.

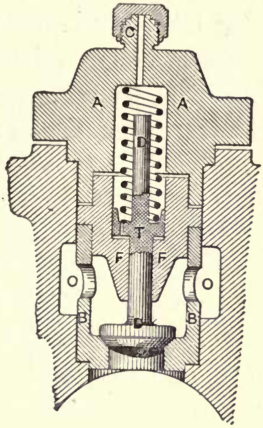


FIG. 63.

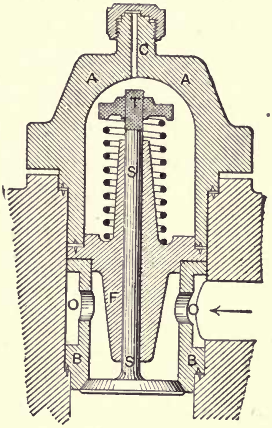


FIG. 64.

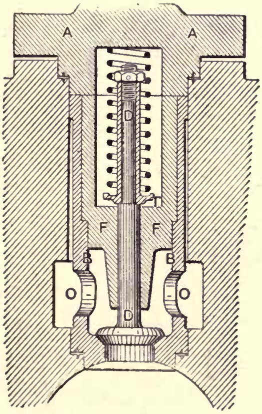


FIG. 65.

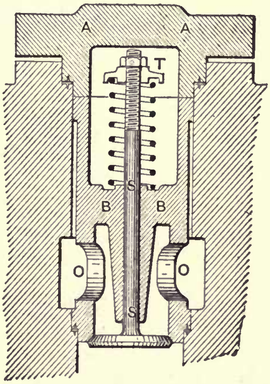


FIG. 66.

The valves of Figs. 65 and 66, in which the plate *T* which takes up the spring pressure is not made in one piece with the rod but is screwed on (and secured by a pin), are somewhat simpler in construction. In particular, it is here no longer necessary to make the guide-box *F* in two parts, but this can then be made in one piece with the casing made as described above for the suction-valve of Fig. 64. The length of the guide offers no difficulties here, but there is the advantage that the tension of the spring can be regulated by varying the adjustment of the plate *T*. Whether this advantage is not more than counterbalanced by the danger of the automatic separation of the plate *T* from the rod owing to the pin falling out, is as yet uncertain. Since the valve-casing *B* is here fairly long, it is not made to touch the wall of the valve-chamber along its whole length, but is provided with two fitting strips at the two ends. The removal of the chamber is also greatly facilitated in this way. In order to be able to do this readily, it is convenient to provide the upper part with a thread into which can be screwed a suitable piece of gas-pipe by means of which the removal can be effected (Fig. 67).

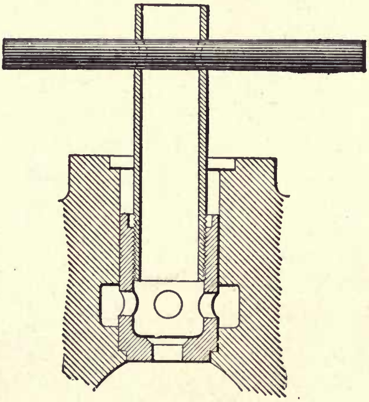


Fig. 67.

As regards the *material*, the valves with their rods always consist of the finest crucible steel, and the guides and chamber, with the seats on which the valves have to slide smoothly and with a sufficiently broad surface, of phosphor-bronze. In hot compression processes, owing to the unequal expansion of the guide and rod, the latter occasionally jams. This evil can be entirely removed only by making the guide of cast-iron. The cover *A* is mostly made of the same fine-grained cast-iron as the cylinder, but sometimes, for the sake of appearances, of bronze. For the valves to work accurately, it is necessary that all the parts belonging to a valve shall be fitted together as carefully as possible before it is introduced; especially to be recommended is the grinding on the seat of the valve when fitted up. So also, in some cases, is the after-grinding by the engineer.

The diameter of the cylinder amounts to from one-third to one-fourth of the stroke, according to the size of the machine, the mean piston velocity not exceeding 2 feet per second. The back suction-valve may have an opening equal to half the surface of the piston, while the one in front scarcely has one-fourth of this value, so that the mean velocity on suction varies between 4 and 8 feet per second. The pressure-valves on both sides have the same dimensions, the opening being one-seventh to one-tenth of the piston surface; the velocity of the carbon dioxide on expulsion can therefore attain to 30 feet per second, but on the average scarcely exceeds 12 feet.

18. Regulation and Safety Arrangements of Carbon Dioxide Machines.—Since carbon dioxide is absorbed only to a slight extent by lubricating materials, of which glycerine has yielded the best results, and since, further, when carried into the apparatus, glycerine does not impede the passage of heat through the heating surfaces, the use of separators is here unnecessary. The machines have, therefore, a simpler external appearance than the ammonia machines.

The *regulating-valve* usually resembles in construction the shut-off valve of the machine. As is shown by Fig. 68, it always consists of a casing *G*, generally of cast-iron, from the top of which the valve-rod *S* projects through a stuffing-box. This rod, which can be moved up and down by means of a fine screw in the case, carries the valve-cone on a rotatable cap *K*, which can be fixed by means of a pin. Finally this cap presses on the bronze valve-seat *L*, which is screwed into the case from below and held perfectly secure in its position by the flange *F*. The cap *K* is required, owing to the fact that the spindle *S* and the seat *L* have different threads and so render exact centring impossible. These valves have been found to remain very tight so long as the surface of the cap is in good condition; if, however, this once becomes attacked (as may readily happen from traces of hydrochloric acid in chemically prepared carbon dioxide—not obtained from natural sources), it must either be turned down again or, what is better, replaced by a new one. After the working season the engineer should not fail to open all the valves and smear such caps as are in good condition with fat; this must be removed before using them again.

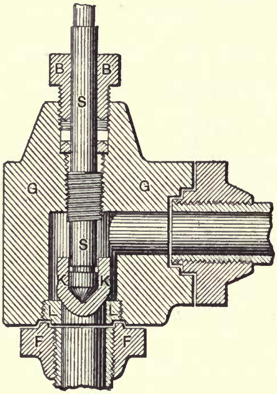


FIG. 68.

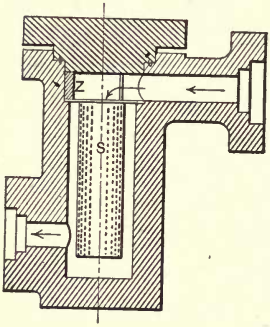


FIG. 69.

In order to prevent solid bodies (solder, packing, filings, or scale) obtaining access to the compressor cylinder, it is the universal custom to insert in the suction-pipe *dirt-catchers* similar to those employed with ammonia machines. Fig. 69 shows one of these, containing, in a closed, strong-walled pot, a tube of sheet metal through which the carbon dioxide passes from the top to the bottom. The sieve with the mud collected in it can be removed after the cover is taken off. The sieve is frequently taken completely away—after it has been ascertained that the tubes contain no further solid matter—in order to diminish the resistance offered to the passage of the gas.

Finally, we must mention the *safety arrangements* for preventing too great a rise in the pressure, which might cause an explosion of the cylinder. The danger is not so great in the case of the tube-systems, because, when the pressure is too high, the tubes only tear and empty with a great noise, without pieces of metal being blown about. Furthermore, abnormal pressures in this apparatus are shown by the manometers, by the readings of which the regular working of the machine can always be controlled. This is scarcely possible in the compressor, and nearly all the accidents are due to the machine being started before the pressure shut-off is opened. The carbon dioxide drawn in through the compressor is then forced into the narrow pressure-channel of the cylinder, where it soon develops an enormous pressure, so that if there is no opportunity for the gas to escape the cylinder must inevitably burst with explosive violence.[1] It is therefore necessary to furnish the pressure channel with a safety arrangement. This may have one of three forms, according as, in case of excessive pressure, the carbon dioxide is allowed to escape from the pressure channel into the air or into the condenser or into the evaporator. The first method is exactly analogous to the use of safety-valves on steam-boilers; it has the advantage of warning the engineer of the danger by the sudden loud noise emitted. But it always means a loss of a portion of the carbon dioxide. The safety-valves with spring loading which are made use of are of the ordinary construction; they generally blow off at 150 atmospheres and have the disadvantages of easily becoming jammed and unreliable after some time. On this account it is often preferred to replace the valve-cone and spring by a thin cast-iron plate which immediately bursts under a pressure of 150 atmospheres and leaves the whole opening free. Such a valve is represented in Fig. 70. On the plate *P* is screwed a cylindrical tube, closed at the outer end and having side openings in it; this catches the fragments of the plate and prevents them from flying about. When the safety-valve comes into operation, the first thing to do is to stop the machine; only after this is done should other valves be opened or closed.

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[1] With good, fine-grained cast-iron and the powerful construction now general, this occurs with an internal pressure of 7000–8500 lbs. per square inch, so that the factor of safety of the cylinder walls under the normal working pressures of the condenser (roughly 80 lbs. per square inch) is about 8–10; these may hence be regarded as quite safe. Further, it can easily be calculated how many rotations are required to cause fracture when the safety arrangements are not acting. The mean stroke-volume of a No. V carbon dioxide compressor is about 0·17 cubic foot and the suction pressure 400 lbs per square inch (absolute), so that for the first rotation the volume is roughly 0·34 cubic foot, which is forced into the pressure-channel—in the more recent machines, cast in one piece with the compressor cylinder—having a volume of 0·035 cubic foot at the most; the pressure there will hence rise to nine times the suction pressure, i.e., to 3600 lbs. per square inch, if the channel were previously empty. Each succeeding rotation approximately doubles this pressure; for although, on the one hand, as the end pressure becomes greater, the increasing back expansion from the clearance diminishes the effective weight of gas drawn in, yet, on the other, the absence of any means for the heat to escape causes the pressure to rise more rapidly than is here assumed. In any case, after the second rotation the pressure exceeds 7000 lbs. per square inch, which is near the breaking limit.

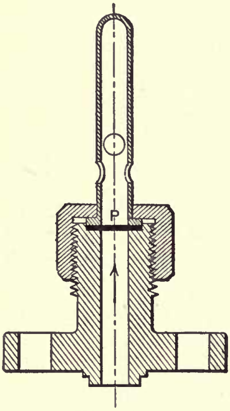

FIG. 70.

For *packing* the joints in the carbon dioxide tube, *lead* is employed where there is no appreciable shaking and in other cases, "*Vulcan*" fibre, which is a mass of several layers of prepared paper pressed together and made uniform hydraulically and may be bought in the form of plates.[1]

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[1] The various proposals which have been made to improve the working of carbon dioxide machines, e.g., by a feed-cylinder, intermediate vessel, etc., will not be considered here, as they have not yet emerged from the experimental stage.

19. Sulphur Dioxide Machines, as we have already seen, are distinguished from the two previous systems by relatively *large cylinder volumes* and *low pressures.* From a purely theoretical point of view they approach the more nearly to ammonia machines; like carbon dioxide machines they appear, however, more simple than these, owing to the omission of oil-separators and rectification apparatus.

Characteristic of sulphur dioxide is the fact that in consequence of condensation in the compressor, it becomes heated to a greater extent [1] than carbon dioxide or ammonia—with the same condenser and evaporator temperatures—and it is on this account that the *compressor cylinders* of sulphur dioxide machines are provided with *cooling* arrangements (see Figs. 71–73).

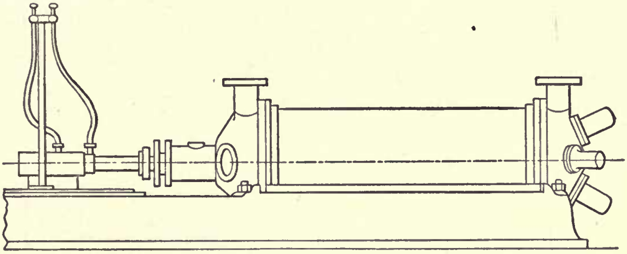

FIG. 71.

This consists of an annular jacket which surrounds the real cylinder and has cold water flowing through it. Further, the *piston-rod* is made hollow and has water led into it at the cross-head (Figs. 71 and 74); the water passes then into the piston, which is also hollow, and thence it is carried back through a narrow, thin-walled tube to the cross-head, where it escapes. Since the cross-head participates in the motion of the piston-rod, the entry and exit tubes for the water must also be movable. A good arrangement is to join the two tubulures at the cross-head by long rubber tubes with the fixed water-pipes, the cocks of which are placed on a frame above the middle of the cross-head path.

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[1] This rise of temperature depends essentially on the ratio between the absolute condenser and evaporator pressures, the values for this (see Table III, page 30) for temperatures of 68° and 14° being SO_2, 3·23; NH_3, 3·01; CO_2, 2·14.

This method of introducing and withdrawing the water is inelegant from an engineering point of view, but it is the most convenient and also the safest. Otherwise the water connections with the cross-head must be made with tubes which are capable of sliding one inside the other so as to change the length and which must join up at both ends with rotary branches. That this method might lead to numerous leakages, and hence to irregular working, is evident. The use of the elongated piston-rod as a pressure-pump has also not persisted, since it is apparently the desire to keep the compressor as far as possible accessible and free from all complications. The cooling of the compressor cylinder has its greatest value in protecting the packing of the stuffing-box, and is of comparatively little use in improving the working of the machine, as is readily seen from the fact that the temperature of the water in the mantle and in the hollow piston-rod is always higher than that of the sulphur dioxide vapours drawn in. These must, therefore, be heated up to the temperature of the water during the compression before they can give heat up to it. During the suction period and the first half of the period of compression, the jacket water serves mainly for cooling the previously heated cylinder walls; it is, indeed, not impossible that during this time heat may be imparted to

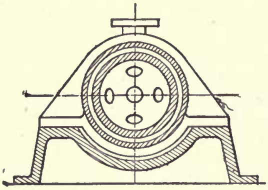
Fig. 72.

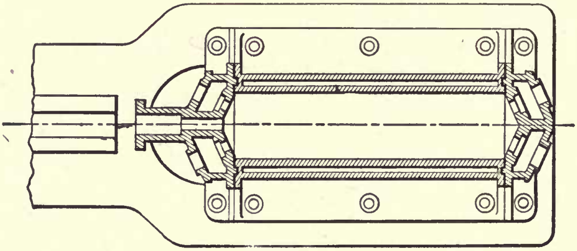
Fig. 73.

the sulphur dioxide, so that the final temperature of superheating may be raised.[1] In the more recent machines the cooling is often limited to the stuffing-box and jacket, without any disadvantage being met with on account of the piston-rod and piston not being cooled.

Owing to the dimensions of the cylinder—the external diameter of which is further increased by the cooling-jacket—arranging it on a base or fixing it to a side-frame would increase the weight of the machine too much, so that it is preferable, as indicated in Figs. 71–73, to place the cylinder partly on a trough of the bed or on two supporting side-brackets and to screw it on by means of lateral flanges strengthened by powerful ribs.

The *stuffing-boxes* of sulphur dioxide machines present, as a rule, little of note. A new arrangement without cooling is shown in Fig. 74, from which can be seen both the method of fitting the hollow piston-rod to the cross-head and also the arrangement of the water circulation inside the rod. Just as is the case in ammonia and carbon dioxide machines, the stuffing-box is provided with a lantern under the suction pressure; behind this, towards the cylinder, is an elastic packing

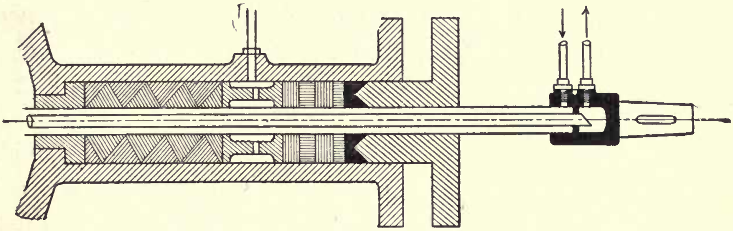

FIG. 74.

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[1] The heat taken up by the water circulating in the jacket and piston-rod appears, according to the researches of *Gutermuth* (Zeitschr. d. Vereins d. Ingenieure, 1889, page 290) and of the *Munich Commission* (*Schröter*, Untersuchungen an Kältemaschinen verschiedener Systeme, Munich, 1890), to be influenced but little by the quantity of water, which becomes heated considerably when the circulation is slow, but only to a slight extent when it is rapid. The ratio it bears to the condenser yield varies, on the average, between 3 and 10 per cent, according to the condition of the sulphur dioxide drawn in and its final pressure.

of metal rings, and in front of it a little mica packing on which the gland presses by means of a rubber ring. Since, with moderately low evaporator temperatures, the suction pressures of these machines fall below the external atmospheric pressure, there is some danger of air penetrating through the stuffing-box into the machine and so exerting a disturbing influence. We should therefore recommend that the lantern of the stuffing-box be connected either with the pressure-pipe or, as *Pictet* has recently done in the compressors of his laboratory[1] in Berlin, with a separate vessel of sulphur dioxide. With the moderate pressure of the sulphur dioxide in the condenser or in this separate vessel no great loss of the gas need be feared if the stuffing-box is kept in fairly good order. With superior construction the lantern may be entirely dispensed with and stuffing-boxes used which do not differ from those used with steam-cylinders. In order to keep them quite cool, it is sufficient either to displace the neck of the stuffing-box entirely into the front suction space (Fig. 75) or to surround it with a cooling-jacket. Both of these arrangements have given good results in practice.

On account of the oily consistency of liquid sulphur dioxide, continuous *lubrication* of the stuffing-box and piston is generally not attempted, and indeed the box would have to be kept cool for this to be done; when a stuffing-box occasionally heats, the engineer often makes use of a little solid fat or rubs the piston-rod with machine fat. From this it follows, as already mentioned, that sulphur dioxide machines require no arrangement for the separation of the lubricating material.

The traces of fat which are carried from the stuffing-box into the interior of the cylinder are very useful for the lubrication of the valve-spindle, but care must be taken that this fat does not solidify there and so fix the suction-valves. This trouble often occurs if the compressor draws in wet vapours at a low evaporator temperature. There then takes place, in the valve-chamber, evaporation of the particles of liquid carried there, this being accompanied by an intense cooling action which leads to the solidification of the fat. In this fact is to be sought the reason for the dry compression process, which has been preferred from the first for sulphur dioxide machines and which is also justified economically, as we have already seen.

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[1] See "Mitteilungen aus dem Institute R. Pictet," by M. *Altschul*, in Zeitschrift f. d. ges. Kälte-Industrie, 1895, Heft 11 and 12.

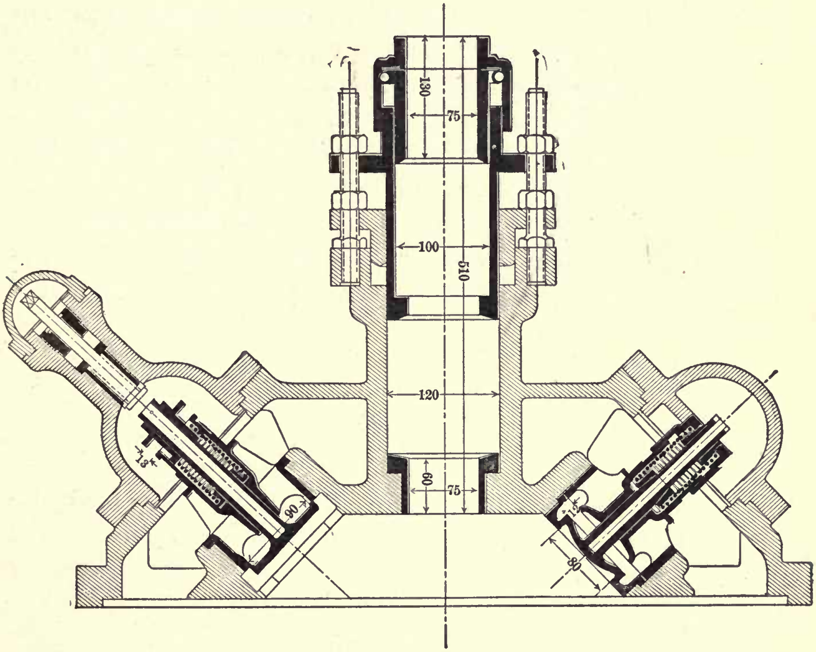

Fig. 75.

The *valves* (usually four on either side) are, in the later machines—e.g., those of Messrs. *Borsig*—made almost exactly like those of ammonia machines, and are built in the cylinder-cover, which consists of several chambers.

The valves of Messrs. *Schüchtermann & Kremer*, of Dortmund, which are represented in Fig. 75 and are evidently fashioned on the plan of the older Pictet's construction by enlarging the guides, have plane seatings.

To prevent the suction-valve falling into the cylinder owing to the rod breaking, a cross-piece is placed under it in the box, and this forms at the same time the limit of the stroke. The whole construction of the valve conditions a not inconsiderable increase of the clearance, which is, however, of little importance with the large cylinder volumes of machines of this type.

Very noticeable in all sulphur dioxide machines is the noisy action of the valves, which is to be ascribed mostly to the absence of liquid lubricant from the valve-seat. Since this phenomenon, which has been found to have no harmful effect on the life of the valve, occurs in exactly the same form with air-compressors, the superheating of the vapours, by which they more nearly approach the true gaseous form, must here exert considerable influence. On account of the more economical working of superheating compared with the so-called wet compressor process, it seems quite right to put up with the noise of the valves.

The construction of the valves in large cavities of the cover, shown in Fig. 75, has proved advantageous, inasmuch as these cavities possess the equalising action of an air-vessel.

Owing to the relatively small density of sulphurous acid, the *piston velocity* may be increased to 5 ft. per second and the cylinder diameter preferably lowered to barely half the stroke. The area of valve-opening is taken as $\frac{1}{10}$ to $\frac{1}{15}$ of the piston area, by which mean velocities of 72 ft. or higher are attained.

Beyond this the compressors of sulphur dioxide machines present nothing worthy of note.

20. Testing of Compressors.—To judge of the efficiency of the separate parts of the compressor of a cooling-machine it is necessary to investigate it with the help of the *indicator*. Such investigation should be made with every machine shortly after its erection and, subsequently, at least once every two years at the end of the working season. For this purpose all good makers provide their compressor cylinders with indicator supports, which are bored along their whole length to the interior of the cylinder and are generally locked by a screw or flange (see Figs. 76 and

77), besides which they are provided with a thread to take the indicator cock. It is greatly to be deplored that up to the present no definite standards for these threads and connections have been agreed upon, so that in almost all cases intermediate pieces have to be procured before the investigation. This presents difficulties, especially with carbon dioxide machines, where, in order not to increase the clearance, the indicator opening is closed by a pin which is not easy to remove, owing to its length. It would be better if it were resolved to furnish the opening, first of all, with a flange (as in Fig. 77), carrying both the thread for receiving the cock and the bored pin for filling the clearance.

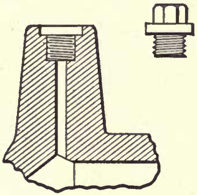

Fig. 76.

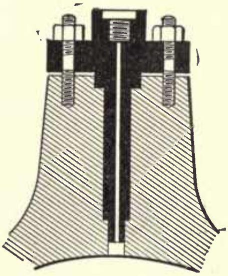

Fig. 77.

As indicators[1] for sulphur dioxide machines those used with steam-engines may be employed, but for ammonia machines it is necessary to use special instruments, made entirely of iron and steel and fitted up just like ordinary indicators.[2] The diaphragm indicators, owing to the irregular bending of the diaphragm, have not given good results compared with the piston instruments, in spite of the fact that they let no ammonia escape.

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[1] On the fitting up and management of indicators, see P. H. *Rosenkranz*, Der Indikator und seine Anwendung, Sixth Edition, Berlin, 1900.

[2] The author has occasionally tested ammonia compressors with ordinary bronze indicators, but always taking the precaution to remove the piston after every reading, to clean it, and replace it again after dipping it in fresh compressor oil. The indicator cylinder is also to be rinsed with fresh Baku oil after each time it is used.

With carbon dioxide machines the pressure becomes so great that one cannot manage with the ordinary indicator pistons, having a diameter of $\frac{3}{4}$ inch, but must employ smaller pistons, 0·25 or 0·4 inch in diameter, working in a lower, narrow part of the indicator or in a box screwed therein (so-called *Riedler pistons*).

The high pressure of the carbon dioxide also renders very difficult the application of the normal indicator cocks, so that it is better to replace these by small screw-down valves (Figs. 78 and 79), which need not have a bore greater than $\frac{1}{6}$ inch, and which have a blow-through plug, *S*, fitted to the cavity in connection with the indicator, in order to restore the atmospheric pressure under the indicator piston after the reading.

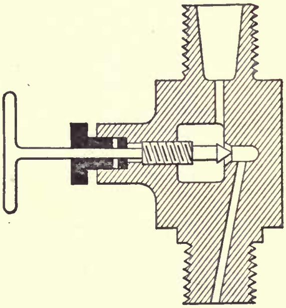

FIG. 78.

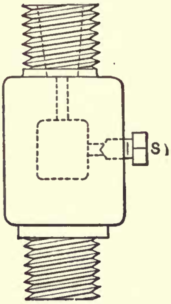

FIG. 79.

Before the indicator cock is fixed to the compressor, the suction-pipe is closed and the machine allowed to make a few revolutions, by which means the cylinder contents are, for the most part, introduced into the pressure-pipe and a lower pressure produced in the cylinder.[1] The machine is then stopped, the pressure pipe closed, and the cocks or valves screwed on. The air which has entered the cylinder must finally be expelled by opening the cocks and, to a slight extent, the suction-pipe, after which the pressure-pipe is opened again and the machine started. As soon as the suction-pipe is quite open, the indicator may be used. On the indicator diagrams obtained, the straight lines corresponding with the condenser and evaporator pressures must be drawn in as soon as possible, for which purpose the manometer must be previously compared with the indicator.

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[1] With ammonia machines the gas remaining in the cylinder is absorbed by water.

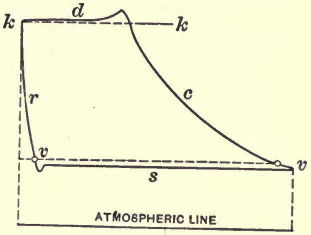

Fig. 80.

If all parts of the machine are in order, diagrams like that in Fig. 80 are obtained, the suction line *s* lying a little below the evaporator pressure *vv* and the pressure line somewhat above the condenser pressure *kk*. The work required for opening the two valves is shown by two little peaks, one on the suction line pointing downwards and the other, on the pressure line, upwards. The influence of the clearance is seen from the back expansion curve *r*, which cuts the evaporator line *vv* only after the change of stroke and so diminishes the actual suction volume, quite apart from the fact that the vapours are taken from the compressor, not with the evaporator pressure, but with the somewhat lower suction pressure. This has for a consequence that the compression curve *c* in the diagram reaches the evaporator pressure only after the change of stroke, so that, instead of the theoretical volume indicated by the length of *vv*, the compressor has a smaller suction volume for vapours with the evaporator pressure. To determine the indicated work, the area enclosed by the curve is determined for a number of diagrams (taken at intervals of from 10 to 30 minutes), either by the planimeter or by dividing the diagram into ten vertical strips and dividing the sum of the mean lengths of these by 10, this giving the mean height of the diagram. Since for every indicator it is known how many inches correspond with a pressure of 1 lb. per square inch, the mean indicated pressure is known at once, and this multiplied by the area of the piston in square inches gives the mean piston pressure in pounds. As the piston-rod makes the areas of the two faces of the piston different, this process must be carried out for both sides. If then the mean value of the two piston pressures is multiplied by the path described by the piston in 1 second—i.e., by the product of double the stroke and the revolutions per second—the work in foot-pounds is obtained, and this, divided by 550, gives the horse-power.

If, for example, we consider an ammonia compressor of 10 inches diameter with a piston-rod 2 inches thick and a stroke of 15 inches = 1·25 feet, then

The *piston surface* is 75·40 sq. in. in front and 78·54 sq. in. behind. Further, if the diagram gives

The *mean indicated pressure* as 41·54 lbs. per sq. in. in front and 42·80 lbs. per sq. in. behind, then

The *piston pressure* will be 3133 lbs. in front and 3361 behind, the mean being 3247 lbs.

If the compressor completes 60 revolutions per minute, the path described by the piston is 2×1·25 = 2·5 feet per second, and the work done 3247×2·5 = 8117·5 ft.-lbs. per second, which corresponds with 14·76 horse-power.

The indicator diagram is also useful in pointing out errors in the compressor. Thus, *too large a clearance* flattens the back expansion curve (see Fig. 81) and means that the suction volume is being made insufficient use of. A similar effect is produced by the *jamming open of the pressure-valve* (Fig. 82) by the spindle taking up an oblique position in its guide. This defect,

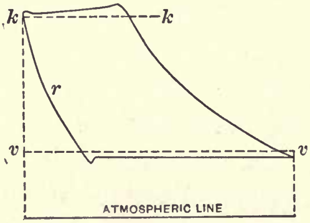

FIG. 81.

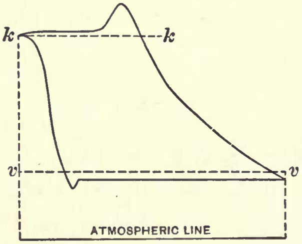

FIG. 82.

which is often introduced after long working, frequently causes also a somewhat late opening of the pressure-valve and a large expenditure of work (large peak in the pressure curve). Since, after the opening, the loose valve immediately fixes itself wide open, the subsequent course of the pressure curve is favourable, and only when the valve shuts at the instant of the change of stroke does the defect come to light again.

With a long stroke, the pressure-valve may also *rebound from its seat.* This is shown by a hook in the back expansion curve (Fig. 83), which runs very level from this place and has as a consequence a very insufficient using up of the stroke volume.

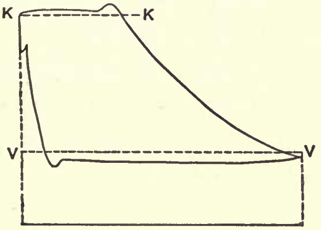

Fig. 83.

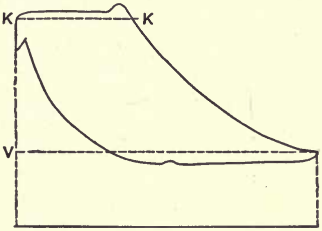

Fig. 84.

If, as in Fig. 84, the back expansion curve proceeds normally after the hook, there is nothing wrong with the play of the valves, but only a *clutching of the indicator piston or vibration of the indicator spring.*

Considerable losses of work arise also by *too high resistance in the suction- and pressure-pipes, too strong a loading of the valves or faulty adjustment of these or the presence of an air-cushion* in them, which are indicated by too great a distance of the curves concerned from the evaporator- or condenser-pressure. In such case the valve-springs must first be replaced by weaker ones or more play must be given to the air-buffer, and only when no appreciable improvement is obtained in this way should constrictions (not, of course, by an insufficient opening of the suction- and pressure-valves) in the pipes be looked for. These should then be immediately removed.

Jamming of the suction-valve in the closed position will cause a considerable fall of pressure before the commencement of suction and the formation of a large peak in the suction curve (Fig. 86), whilst, if it jams open, the compression is greatly delayed; i.e., part of the stroke of the piston is wasted.

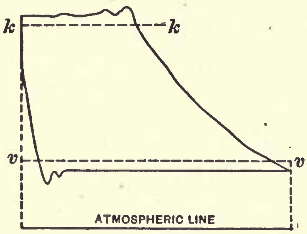

Fig. 85.

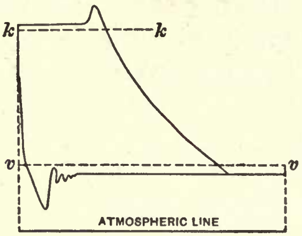

Fig. 86.

Much looseness of the valves is shown (Fig. 87) by the disappearance of the horns and a gradual transition of the compression curve into the pressure curve and of the back expansion into the suction curve; further, the compression curve is almost rectilinear and steep when the pressure-valve is the looser and flat when the suction-valve is the looser.

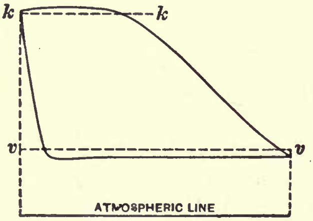

Fig. 87.

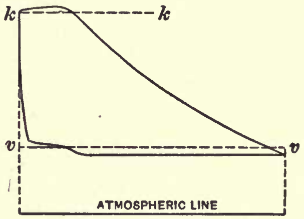

Fig. 88.

When the piston does not fit tightly the gas passes from one side to the other, i.e., the pressure rises too slowly on the compression side, and the compression curve becomes very flat (Fig. 88). The late, and therefore sudden, opening of the pressure-valve in this case is immediately recognisable by the vigorous banging of the valve. On the other hand, the pressure on the suction side, when the piston changes its direction, easily remains higher than that prevailing in the evaporator and so prevents the opening of the suction-valve. This lasts until the piston velocity has become so great that the quantity creeping past the piston is no longer able to fill the suction space. The pressure there then gradually falls and the suction-valve begins to work.

If several of the above-mentioned defects are present, very involved diagrams may be produced, but their interpretation should offer no great difficulty.

It may be useful to remark here that regular oscillations of the various curves about their normal course are always to be ascribed to vibrations of the indicator spring and are not to be put down to any action of the valves.

CHAPTER IV.

APPARATUS FOR GIVING OUT AND TAKING UP HEAT.

21. **The Action of the Condensers** of cooling-machines consists in the giving up of the heat taken up in the evaporator, together with the equivalent of the compressor work, to water which thus becomes heated or, if it absorbs the heat at constant temperature, evaporated. Not infrequently these two processes are combined, so that only a part of the cooling-water reaches a higher temperature, the rest being evaporated; or it may happen that the previously heated water, owing to subsequent partial evaporation, falls to its original temperature. Under no circumstances should the water come into direct contact with the cooling-agent, so that the injection condensers used with steam-engines are here excluded and only the so-called *surface condensers* can be employed.

The vapour coming from the compressor generally enters the condenser in a *superheated condition,* and is there first *cooled to its saturation temperature,* at which it is *liquefied,* the liquid being then *cooled down* to a somewhat lower temperature (if possible, nearly to the initial temperature of the water itself). So that in the condenser there take place at nearly constant pressure three distinct heat transferences, that due to the liquefaction being, under normal conditions, the most considerable. If we take only this into account at first, it is evident that if the cooling-water is only heated and not evaporated, its final temperature will be at most the condensation temperature of the cooling-agent. With a suitable arrangement these two temperatures will, indeed, nearly coincide. They will rise together as the initial temperature of the condenser water becomes higher and as the quantity of the latter diminishes relatively to the refrigerating capacity of the machine. In order to permit of a control of this important factor in the working of the machine, we give in Table X the approximate final temperatures of the cooling-water for different initial temperatures, calculated[1] for an hourly cold production of 400,000 B.T.U. at an evaporator temperature of 14°.

The raising of the condenser temperature produces, as is known, an increase of the pressure and of the amount of work required; this can be arrived at, with the help of vapour-pressure tables for the various cooling-agents, but not from the following tables, by subtracting the cold produced from the condenser yield, since the values given for the latter must be regarded only as mean estimated numbers. On the other hand, the tables given below allow us to judge beforehand of the advantage of a certain amount of supercooling in front of the regulating-valve, this being produced if the difference between the initial and final temperatures of the condenser water is greater than 18° for sulphur dioxide or ammonia machines or greater than 9° for carbon dioxide machines. Further, it can at once be seen whether the condenser water reaches too high a temperature, and if so, partial evaporation should be made use of. The final temperatures admissible, when the water does not evaporate, are given in heavy figures in the table.

TABLE X.—FINAL CONDENSER-WATER TEMPERATURES.

Initial Temperature.	Quantity of Water in Cubic Feet per Hour.				
	175	350	525	700	875
41° F.	82°	**63°**	**54°**	**52°**	**50°**
50	93	**72**	**64**	**61**	**59**
59	102	79	**72**	**70**	**68**
68	111	90	82	**79**	**77**
77	122	100	93	90	86

It is far simpler to ascertain the water required when pure *evaporation* is employed, since every pound of water absorbs roughly 1080 B.T.U. The temperature of liquefaction of the cooling-agent will, in this case, always lie a few degrees above the dew-point of the air, to which the cooling-water must be raised in the condenser-tubes before it evaporates—if its temperature is not initially higher. Further, evaporation only takes place when the air is not saturated with water-vapour.[1] This being the case, it will go on if the water on the tubes has assumed a higher temperature than that of the air, and help in this direction is obtained by employing the cold cooling-water for lowering the temperature of the liquefied cooling-agent in the condenser. It is well known that the working of an evaporation condenser depends in a very high degree on the weather, that it acts well in dry weather and badly when it is damp, although the quantity of water used is very small compared with that required in ordinary water-cooling.

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[1] The method of calculation has been explained on page 186 *et seq.* of the "Zeitschrift für die ges. Kälte-Industrie" for 1897. The values in the table must be regarded as approximate.

An important part is also played by the absorptive capacity of the air, since this determines the quantity of air to be supplied per hour to the condenser. In a perfectly saturated condition, every 1000 cubic feet of air contains a quantity of water-vapour given in Table XI for various temperatures.

TABLE XI.—WATER-VAPOUR CONTAINED IN SATURATED AIR.

Temperature in Deg. F.	Water-vapour in Lbs. per 1000 Cub. Ft.	Temperature in Deg. F.	Water-vapour in Lbs. per 1000 Cub. Ft.	Temperature in Deg. F.	Water-vapour in Lbs. per 1000 Cub. Ft.
32	0·304	52	0·625	72	1·216
34	0·330	54	0·670	74	1·297
36	0·351	56	0·741	76	1·381
38	0·379	58	0·768	78	1·470
40	0·408	60	0·822	80	1·563
42	0·440	62	0·879	82	1·661
44	0·472	64	0·939	84	1·765
46	0·505	66	1·002	86	1·865
48	0·540	68	1·082	88	1·969
50	0·584	70	1·140	90	2·076

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[1] The determining factor for the evaporation is the *difference of the pressures of the water-vapour at the surface of the water and in the air;* if this difference is negative, condensation will occur and heat thus be given up to the surface of the water.

The absolute moisture-content of the air varies only between 60 and 95 per cent of these values, so that the air is capable of taking up more water-vapour formed by evaporation. This absorbing capacity for various temperatures and moisture-contents of 1000 cub. ft. of the air is calculated from the preceding table to have the following values:

TABLE XII.—ABSORPTIVE CAPACITY OF UNSATURATED AIR.

Air Temperature, Degrees F.	Percentage Moisture-content of the Air.			
	60	70	80	90
50	0·234	0·175	0·117	0·058
59	0·318	0·238	0·159	0·080
68	0·433	0·325	0·216	0·108
77	0·570	0·428	0·285	0·143
86	0·746	0·559	0·373	0·187
95	0·986	0·739	0·493	0·246

From this table it follows, for example, that 1000 cub. ft. of air with a moisture-content, determined by the hygrometer, of 70 per cent can take up, at 68° F., 0·325 lb. of water, corresponding with, roughly, 35·1 B.T.U. For a cooling installation with a cold production of 100,000 B.T.U. per hour, the condenser heat under these conditions would be 126,000 B.T.U., the absorption of which would require the evaporation of about 120 lbs. of water, and this in its turn necessitates the supply of at least 400,000 cub. ft. of air per hour. Since, however, complete saturation of the air by the evaporation cannot be counted upon, the amount of air required must be multiplied by about one and a half.

The same holds naturally for the subsequent cooling, by partial evaporation, of the cooling-water used; the influence of warm and damp weather on the evaporation process in practice is somewhat mitigated by combining it with the above-mentioned ordinary water-cooling, using for a cold production of 100,000 B.T.U. per hour, instead of the theoretical 100–140 lbs. of water required by the evaporation process, 4 to 6 times as much, if possible.

After these explanations no difficulty will be found in deciding on a system of condensation, if the meteorological conditions of the locality are known. As an example, the most important data (monthly averages) for Frankfurt-am-Maine are given in Table XIII, from which it is seen that in the months of February, March, October, and November the condenser cooling is best carried out with river-water, which is obtainable in almost unlimited quantities, whilst from April to September it is best to use well-water. Since the dew-point in the hottest months lies above the temperature of the well-water, *cooling by evaporation should be used only as a makeshift when insufficient well-water is available.* In fact, atmospheric condensers always show higher pressures than submerged condensers, even with the same initial and final temperatures. This is probably due to the fact that the transference of heat from the water to the air requires, even when the evaporation is vigorous, a larger surface per time unit than the same transference taking place between the cooling-agent in the condenser-tubes and the surrounding water. This smaller capacity for heat transference can, of course, only be compensated for by a greater fall in temperature, and consequently, with the same water temperature, by a higher condenser pressure. So that the recent tendency towards the use of atmospheric condensers is not usually justified from an economic point of view. If, however, the only cooling-water available either contains

TABLE XIII.

Month.	Air Temperature, Deg. F.	Vapour Pressure in Inches of Mercury.	Dew-point, Deg. F.	Temp. of the Well-water, Deg. F.	Temp. of the River-water, Deg. F.
January........	32·9	0·154	27·9	46·2	35·6
February.......	35·6	0·165	29·7	45·0	36·7
March..........	41·7	0·181	32·0	45·5	41·9
April...........	50·0	0·217	36·5	46·6	50·4
May............	55·4	0·303	45·3	48·4	57·0
June...........	64·4	0·390	52·2	50·5	65·1
July............	67·5	0·437	55·4	52·9	68·7
August.........	66·2	0·417	54·0	54·0	67·8
September......	59·0	0·374	50·9	54·5	62·2
October........	49·1	0·275	42·8	54·1	51·4
November......	39·7	0·220	37·0	51·6	40·8
December......	32·9	0·173	30·7	48·7	36·3

much organic matter or deposits salts (e.g., lime, gypsum, etc.), and so retards the passage of heat through the tube surfaces, the correct thing is to choose the atmospheric apparatus, which can be cleaned readily and without disturbing the working.

22. **Submerged Condensers and Coolers for Liquids.** — In almost all cases condensers are now made of welded wrought-iron tubes having an internal diameter of $\frac{3}{4}-1\frac{1}{2}$ inches, and a thickness of wall varying from $\frac{1}{8}-\frac{1}{4}$ inch, according to the pressure of the cooling-agent. For sulphur dioxide, weaker copper tubes are sometimes found, and of late years bottle-shaped wrought-iron vessels, from which the condensed liquid is removed by means of internal stand-pipes. If the heat of liquefaction is given up to water without counting on the evaporation of the latter, the water surface need not be very great. In this case, the cooling-water generally flows through a more or less cylin-

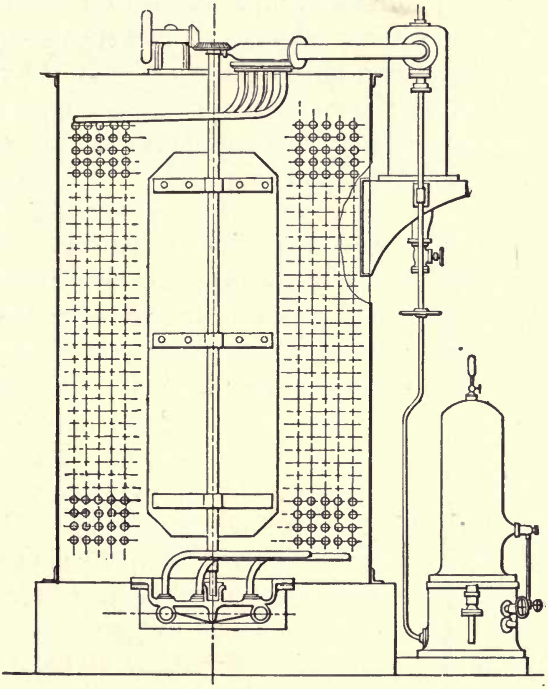

FIG. 89.

drical, open vessel in which the spirally arranged condenser-tubes are submerged, this apparatus being hence known as a *submerged condenser.* The cooling-agent always flows from the top to the bottom of the spiral tubes, whilst the cooling-water passes in the opposite direction and so becomes gradually heated.

In order to ensure the temperature being constant through each horizontal layer of the cooling-water—as required by the principle of counter-currents—and to prevent the water ascending within the spiral without taking up heat, a stirring-apparatus is employed which drives the water towards the outer spirals. Such a condenser, with five systems of tubes, is represented in Fig. 89 [1]; the connections for the entrance- and exit-tubes for the cooling-water are not shown here. The first of these is generally an ordinary socket with a shut-off tap, whilst the escape orifice is mostly constructed as an overflow, as in Fig. 90, and is connected with a graduated cock, *A*, which allows a definite quantity of water to flow through it for a given, indicated head of water. For controlling the head, the overflow chamber is provided with an open gauge-glass, *W*, and to keep the pressure below the cock equal to that of the external atmosphere the cock is joined to the escape-pipe by means of an open funnel.

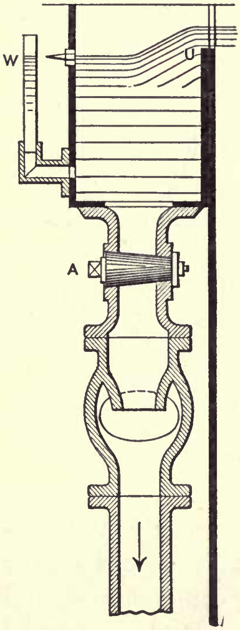

FIG. 90.

The tube-systems of the condenser represented in Fig. 89 all have the same number of coils, the diameters of which, as also the lengths of tube and the cooling-surfaces, increase from the inside to the outside. An increase in the length of tube means also an increase in the resistance to the motion of the cooling-agent, so that, with the above arrangement, the outer tube-systems with larger cooling-surfaces are traversed by a much smaller quantity of the refrigerating agent per unit of time. Further, the water surrounding the outer spirals is much less exposed to the action of the stirring-apparatus, so that the inner tube-systems are of much greater account. The consequence of this is that there is a difference in the excess of temperature of the cooling-agent over that of the water at places at the same level, and hence an unequal transference of heat per square foot of cooling-surface per unit of time.

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[1] This figure shows further the combination of Fixary's oil-separator with the condenser and with the oil-collector below (see Fig. 46).

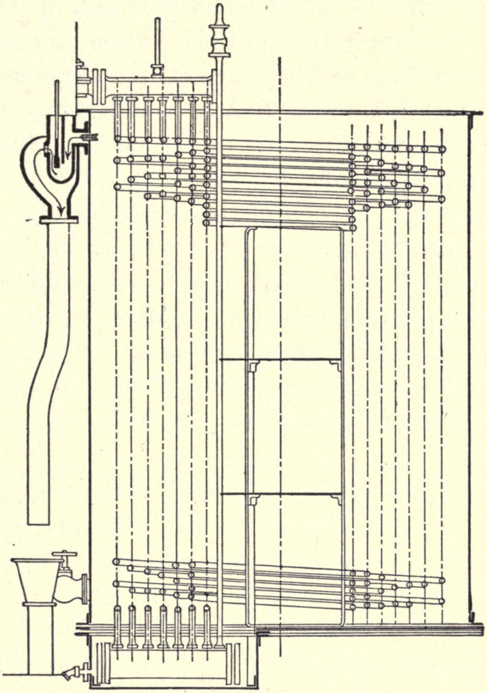

FIG. 91.

This drawback was removed in great part in the *ice-machines of the Linde Company,* where all the separate tube-systems were made to contain the same length of tube. The inner spirals had then, of course, to be given a larger number of coils than the outer ones for the same vertical height (See Fig. 91). But, on the other hand, the separate coils of the inner system came so close together that only a small space remained for the water to circulate, and consequently the action of the stirring-apparatus was diminished. This defect may, however, be remedied by omitting the central stirring-apparatus, by removing the free inner space from the water circulation by a metal cylinder, and by bringing in the water as nearly as possible in a tangential direction, so that it is carried through the coils and rises as a counter-current to the cooling-agent. Further, the introduction of the cooling-water merely into the central cylinder, which is open at the top and communicates only at the bottom with the space containing the coils, has lately been favoured.

In any case, the omission of the stirrer has given especially good results with *liquid-coolers*[1] which, as in Fig. 92, often contain only one coil, but yet allow of a perfect fulfilment of the principle of the counter-current. These are for the purpose of cooling the cooling-agent, after liquefaction, at a high temperature in the condenser, nearly to the initial temperature of the cooling-water, so that it reaches the regulating-valve with diminished heat.

In order to make the liquid flow of itself to this cooler, the latter should be placed lower than the condenser, which makes it necessary to carry the cooling-water through the cooler under pressure, and hence to keep the water perfectly continuous (Fig. 92).

With submerged condensers, the liquid-cooler may, according to the suggestion of *Stetefeld*, be directly united by omitting the stirrer of the condenser and inserting the coils of the liquid-cooler in the free inner space, which must then be shut off from the condenser. The cooling-water then flows in at the top of the inner vessel and enters the outer condenser at the bottom, while the cooling-agent flows in the opposite direction through the coils.

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[1] These are frequently known as *after-coolers* with reference to the refrigerant, or *fore-condensers* with reference to the cooling-water. The above simple expression, which is sufficiently descriptive, should remove all misunderstanding.

The combination in the manner represented diagrammatically in Fig. 93, which is frequently employed, has the advantage of a simple path, *WW*, for the water, but sometimes causes difficulties in the working owing to the formation of bubbles of vapour in the upper tube, *A*, connecting the two apparatus. This occurs when the liquid flows from the condenser nearly boiling and the external temperature is relatively high. The entrance of heat from outside then acts in the same direction as the removal of pressure from the column of liquid at *A*, so that vapour forms there and may prevent the further passage of liquid. This trouble has been met with more especially in sulphur dioxide condensers, in which relatively low pressures prevail, while the column of liquid in the apparatus is a heavy one; by placing the liquid-cooler low down, as recommended above, the difficulty is removed even in this type of machine.

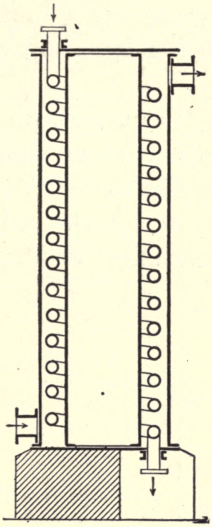

Fig. 92.

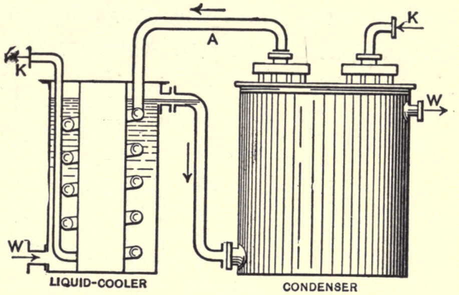

Fig. 93.

A very convenient mode of combining the liquid-cooler with an atmospheric condenser has been lately introduced into the machines made by the *Germania* Company of Chemnitz. Under the trough of the condenser another trough is so arranged that it is accessible and is yet shaded by the first. The lower trough then contains simply a horizontal pipe of the form shown in Fig. 99, and through this the liquefied ammonia circulates, while outside it and in the opposite direction cold water flows round in channels formed by vertical walls. The waste water must finally be transferred to the atmospheric condenser by means of a pump.

A corresponding arrangement for cooling the superheated vapour from the compressor down to the temperature of liquefaction, i.e., for destroying the *superheating* by utilising the waste cooling-water from the condenser, has, so far as I know, only been applied in certain American ammonia machines, although it should, especially with carbon dioxide machines, mean a not inconsiderable saving of water, as the latter could escape with a much higher temperature than it does now, without impairing the condensation.[1] Fig. 94 represents diagrammatically such a

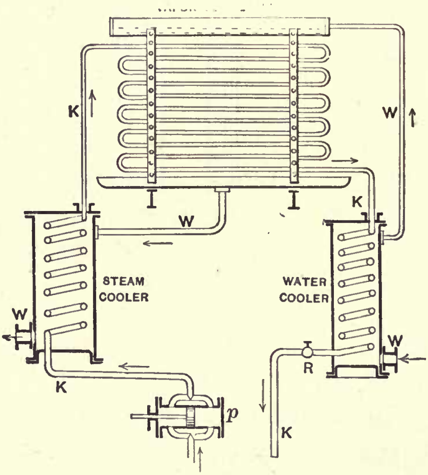

FIG. 94.

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[1] To judge of this, it is necessary to ascertain the quantities of heat cor-

vapour-cooler combined with an evaporation condenser and a liquid-cooler; the water-pipes are denoted by *W* and those for the cooling-agent by *K*, while *R* is the regulating-valve and *P* the compressor; the counter-current principle is in this way adhered to fairly closely.

With sulphur dioxide, the pressures of which render possible condensation in a wide vessel, the counter-current principle may be very exactly conformed to, by carrying the cooling-water, as indicated in Fig. 95, through numerous narrow, thin-walled tubes placed in such a vessel, which is itself surrounded with the water.

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responding with the supercooling, condensation, and superheating of the various cooling-agents and these are given in the following table:

TABLE XIV.

Temperature of liquefaction in the condenser	68° F.			86° F.		
Cooling-agent	NH_3	SO_2	CO_2	NH_3	SO_2	CO_2
Approximate temperature at the end of the compression	158°	149°	115°	208°	190°	150°
Fractional heat of superheating	0·100	0·075	0·228	0·128	0·097	0·405
" " " liquefaction	0·872	0·877	0·639	0·817	0·803	0·346
" " " supercooling	0·028	0·048	0·133	0·055	0·100	0·249

The separate quantities of heat are here given as fractions of the total cooling effect of the condenser, with a supercooling to 50° and supposing the evaporator temperature to be 14°.

From the above numbers it is seen what a large fraction of the total condenser heat is due to supercooling and superheating, especially with carbon dioxide machines, while the heat of liquefaction becomes very small in the neighbourhood of the critical pressure and, beyond this, disappears entirely. Especially do these numbers argue for the separate removal of the superheating in a counter-current cooler. The vapour-cooler required for this purpose is, in America, placed above the submerged condenser (see *Gutermuth*, American Ammonia-Compression Cooling Machines, Zeitschrift d. Vereins d. Ing., 1894, and Zeitschrift f. Kälte-Industrie, 1889, page 46). In the latter place, a liquid-cooler is described in combination with an evaporation condenser, the lower coils serving as a vapour-cooler; this is certainly an arrangement well worth attention.

In Germany, *Sedlacek* has placed the pressure pipe from carbon dioxide compressors in an underground channel, through which the escape-water from the condenser flows; this effects a vapour-cooling, the extent of which is perhaps limited only by the somewhat small cooling surface of the tube.

The objectionable part of this construction is perhaps the large number of joints, which require very careful making and attention.

Concerning the *dimensions of submerged condensers,* experience has shown that, with a mean fall of temperature of from 9° to 13° a transference of heat of 300 to 450 B.T.U. per hour may be reckoned on per square foot of mean heating surface (i.e., the mean of the inner and outer surfaces of the tubes). The thickness of the tubes is of little account, but the proper distribution of the whole surface between the real condenser and the liquid-cooler, according to the quantities of heat to be removed, is important.[1]

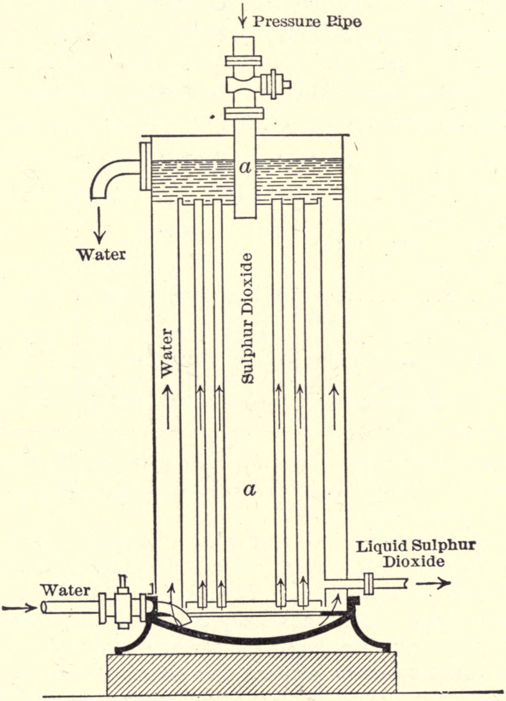

Fig. 95.

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[1] A detailed review of the heat removed during the various stages of the condensation inside the tubes would, owing to our insufficient knowledge of the laws of the conduction of heat, be premature at this time.

23. Atmospheric Condensers and After-cooling Apparatus must first of all give to the water as large a surface of contact with the air as possible; further, they should be erected where there is a strong current of air, so that the air, saturated with moisture by the evaporation, may be quickly replaced by dry air. The first of these conditions is fulfilled by allowing the water of *atmospheric condensers* to trickle over the horizontal tubes in a thin but continuous layer, and the second by erecting the apparatus on the roof of a house or on a high stand (of course, close to the engine-room), the tubes being arranged in the direction of the prevailing winds. If local circumstances do not allow a sufficient natural exchange of air to be expected, it is best to erect the condenser in a closed room and arrange an artificial passage of air by means of ventilators; in large installations enormous quantities of air (see the example at the end of § 21) are necessary, and the work required by the ventilators must not be under-estimated. Consequently they should only be used in time of need, and in all cases the outlet pipe for the warm air (the so-called hood) saturated with moisture, should

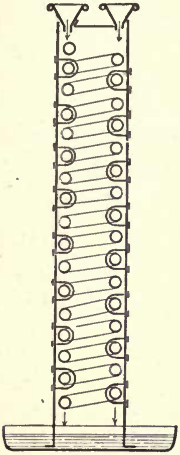

Fig. 96.

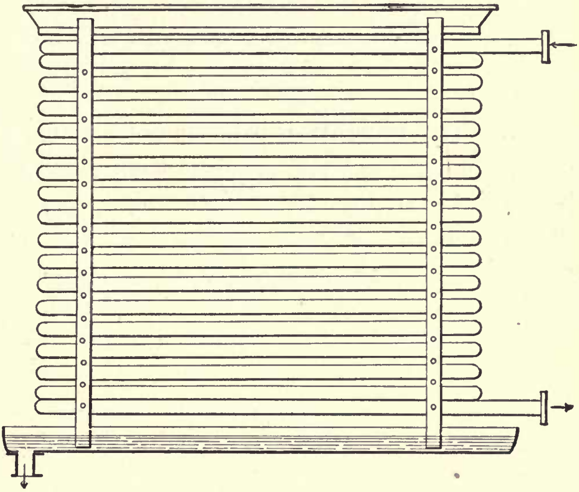

Fig. 97.

be directed vertically upwards, so that its up-draught may be utilised.

In America the horizontal condenser-tubes are joined simply by bends screwed on and tightened with lead, whilst in Germany the tubes are welded together and the ends bent round, just as is done in the case of submerged condensers. In this way are formed systems of tubes like those shown in Figs. 96–98, with free inner spaces which are only imperfectly subjected to the action of the draught; it is hence not possible to make complete use of the tube surfaces lying on the inside of the system, even if the water trickles there in the same way as on the outside. The distance of the separate tubes from one another is not made greater than is required to allow them to be fixed to vertical supports (consisting of flat strips or of angle- or U-shaped iron rods). The whole system, above which stands two open troughs for distributing the water, is arranged in a basin for catching the non-evaporated, excess water, which is either not at all or but slightly warmed and so can be used again immediately.

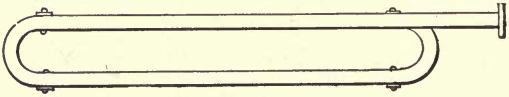

FIG. 98.

The disadvantage which this arrangement possesses, of utilising irregularly the outer surfaces of the tubes for the evaporation, may be avoided by constructing the condenser of tubes bent in the manner indicated in Fig. 99. Professor v. *Linde* suggested that three of these tubes be placed so that the horizontal portions all lie in a vertical plane, in which they are kept by being fixed to vertical supports. In order that the bends of these tubes may not get in the way of one another, one of them must be bent backwards, another forwards, whilst the third is kept in the vertical plane. In

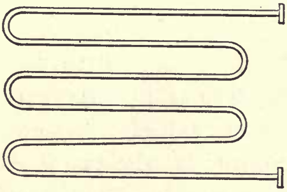

FIG. 99.

this way is formed the condenser element represented in Figs. 100 and 101; it has three entry- and three exit-pipes for the cooling-agent, and these must be joined, either with one another by means of suitable connections or with a collecting vessel. This arrangement presents the further advantage that the outsides of the tubes can be readily cleaned, even during the working. The cold water may here be distributed by means of a trough or by a wider tube furnished with a large number of holes. Perhaps more to be recommended is the use of distributing pipes with *longitudinal slits,* as these can be more easily examined and cleaned. When the water flows out very irregularly, it is usual to close some parts of the slit with wooden plugs, by which means a perfectly regular distribution of the water may be obtained. Such an apparatus always suffers from the defect touched on at the end of § 21—the varying transmission of heat from the interior of the tubes to the trickling water, and from this to the air. To obviate this, many proposals have been made, e.g., the covering of the tubes with cloths or fine wire

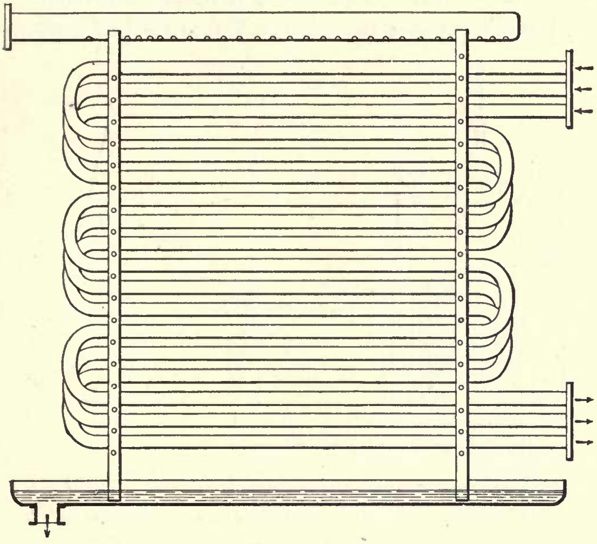

FIG. 100.

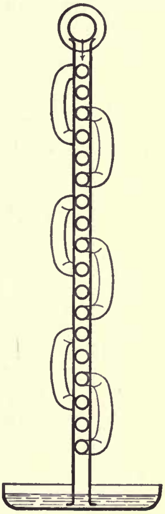

FIG. 101.

gauze, which increase the surface of evaporation. Apart, however, from the fragile nature of these materials, their use does away with the great advantage of the evaporation system namely, its ready supervision; so that when there is great lack of water this evil must be put up with. We shall become acquainted later on, in the after-cooling apparatus, with an excellent means of removing these difficulties, which, however, requires a complete separation of the condenser from the evaporation apparatus.

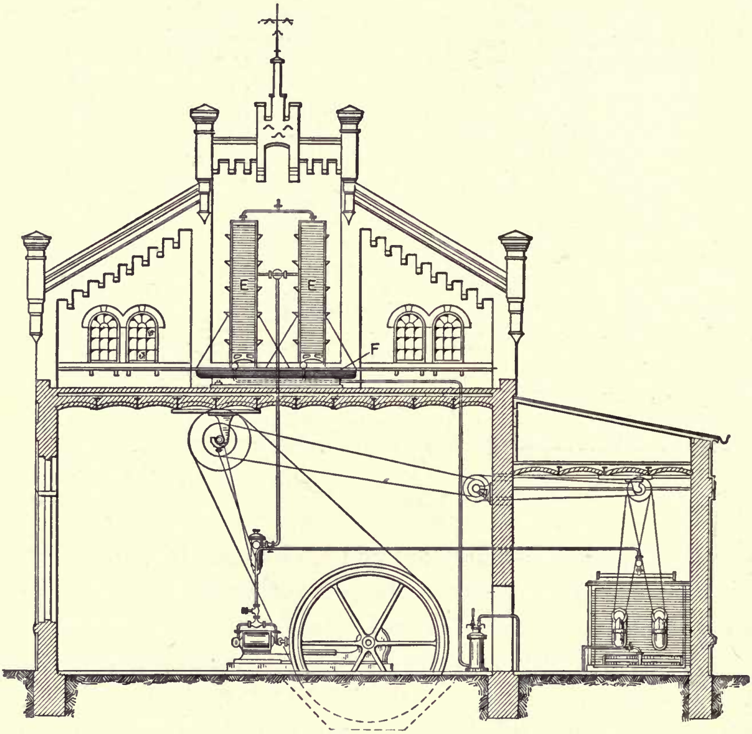

Fig. 102.

The *arrangement of an evaporation condenser* is shown in Figs. 102 and 103.[1] The two apparatus *E*, with the common trough

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[1] Erected by the *Germania* Company of Chemnitz for the Breda Brewery of F. Smits van Waesberghe.

F, are placed directly on the roof of the engine-room and are composed of several superposed tube-systems (Figs. 96–98), which are then joined up to vertical collecting pipes. The water is drawn up from the cold-water tank *H* by a pump, *AB*, and passed through the pipe *CC* into the distributing trough. In order to make use of at least a part of the trickling water carried off as spray by too strong an air-current, at the side of the condenser are arranged a few open troughs, which catch the spray and carry it back to the tubes. The fact must, however, not be disregarded that, with very vigorous aëration, these troughs may

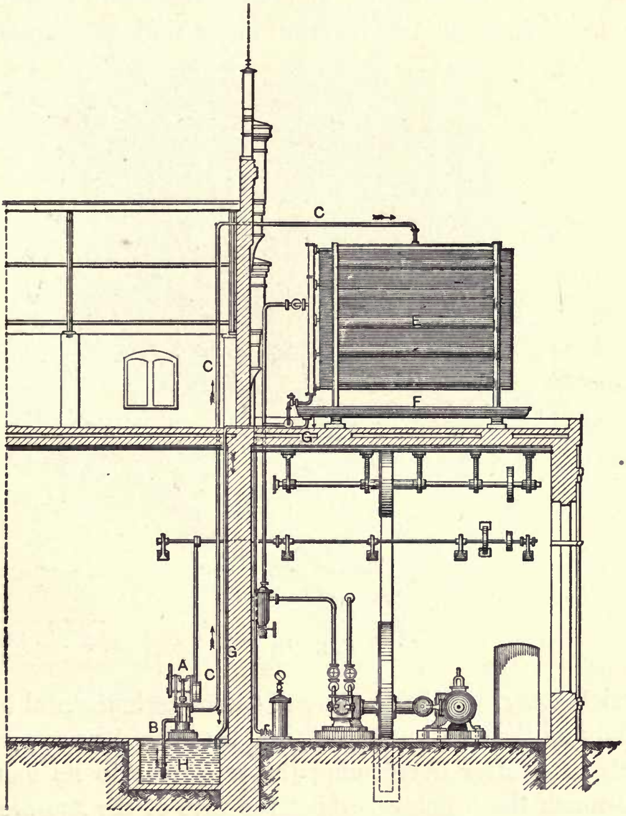

FIG. 103.

have a retarding effect. The non-evaporated water flows from the basin F through the pipe GG back into the tank H.

Although, until recently, any *covering of evaporation condensers* was regarded as a hindrance to the air circulation, it has been found, on more accurate observation, to serve as a protection from the sun's rays. The use of roofs with lateral openings should, however, not be omitted, so that the vapour may be carried away. Suitable also are *louvers* of horizontal boards, especially on the south and west, and these must be so arranged that air can—after overcoming the resistance to its motion—pass through the openings, whilst the rays of the sun are completely excluded. If both the steam-engine and the cooling-machine are supplied with evaporation condensers, these should never be placed close together, otherwise the hot-water vapour from the steam-condenser will impinge on the surface of the condenser of the cooling-machine and impair its action. When there is only little space, the simplest plan is to raise the steam-condenser to a higher level.

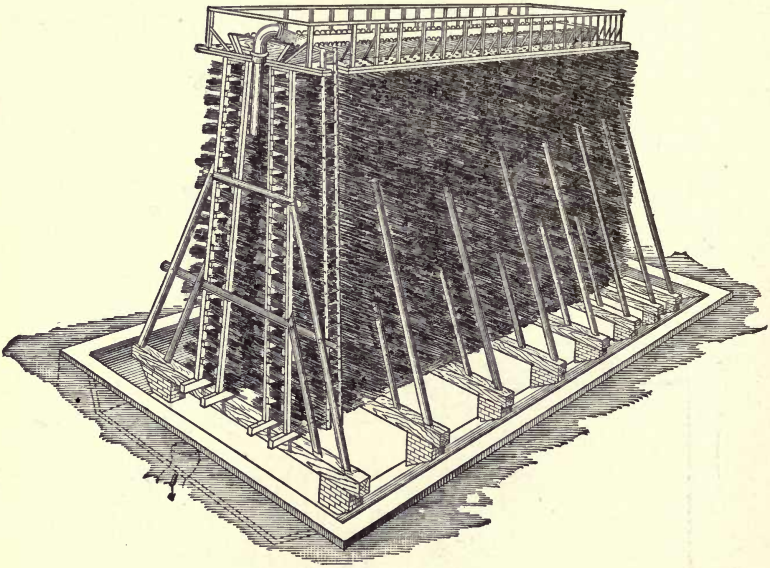

Fig. 104.

The *after-cooling* of the water from submerged condensers consists in restoring it to its original temperature by the evaporation of a small portion of it; for this purpose the whole of the water is *spread out in a shallow layer and exposed to a natural*, or, if necessary, *an artificial, air-current.* This leads, on the one hand, to maximum evaporation, and on the other to regular and rapid distribution of the cooling through the entire mass. The portion evaporated is then to be replaced by cold water and the whole transferred to the condenser. When a liquid cooler is used, the cold water is naturally led to this first, in order to utilise its low temperature to the full.

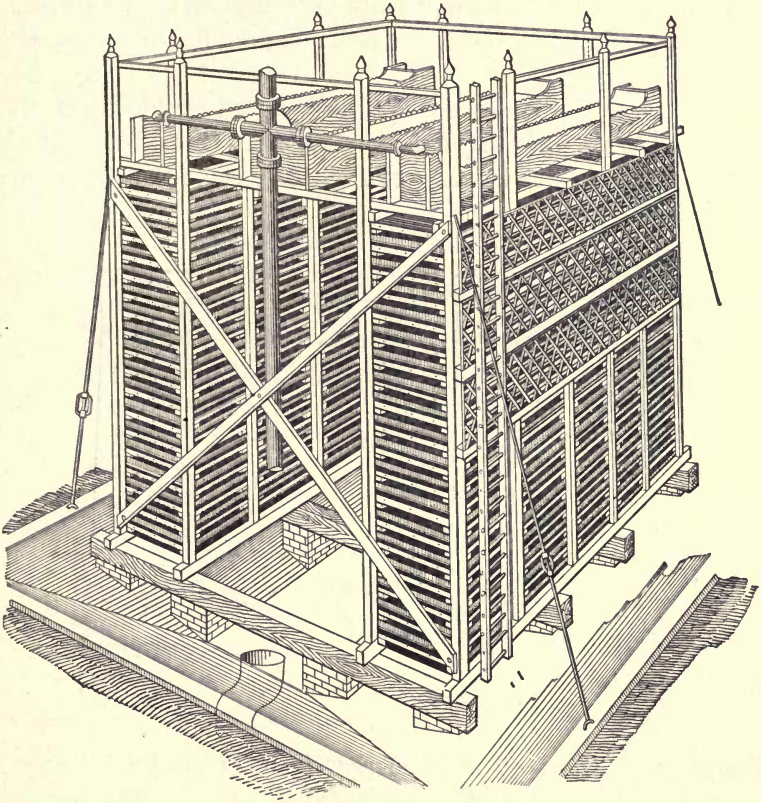

FIG. 105.

The preparation of a large evaporating surface is, without doubt, most simply effected by erecting a frame-work filled with bundles of twigs—a so-called graduation apparatus (used in concentrating brine)—as shown in Fig. 104. The warm water is pumped up into an open trough covering the whole of the apparatus, whence it runs down into the collecting trough through the twigs exposed to the wind, becoming gradually evaporated and cooled meanwhile.

To replace this cheap but not very elegant arrangement, the *Kaiserslautern Wood Company* manufacture structures of notched, horizontal boards or beams (see Fig. 105), while the *Armaturen Company of Frankenthal* (formerly Klein, Schanzlin & Becker) hang, in a tower-shaped space closed in at the sides, narrow, vertical boards arranged in several layers in such a way that the air passing upwards is forced to come into contact as much as possible with the water trickling down. It was formerly thought that such *chamber-coolers* would only act with artificial aëration, but it has recently been shown that, with a suitable arrangement of the surfaces, the warming of the air by the water is generally sufficient to supply air enough to the apparatus. Figs. 106 and

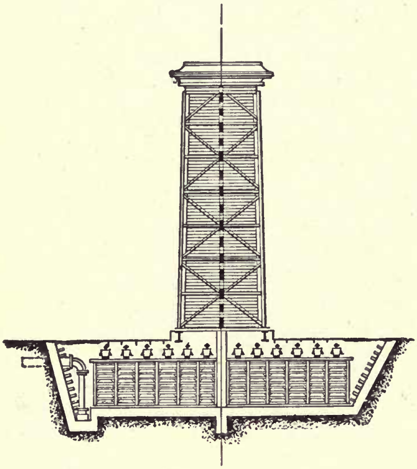

Fig. 106.

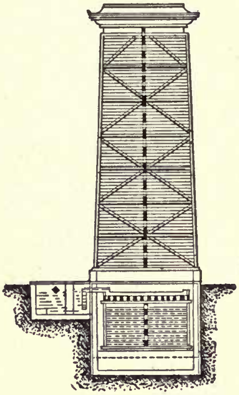

Fig. 107.

107 represent such a chamber-cooler (made by Messrs. *Balcke* & Co. of Bochum) in combination with a condenser. In this so-called underground condenser the convenient separation of the evaporation and condensation processes is especially distinct.

Another after-cooling apparatus worthy of attention is that of *Körting,* in which the warm water is very finely divided by means of a peculiarly shaped jet and is thus given a very large evaporating surface. The latter can be still further increased by catching the spray in a wooden or metal graduator, from which it slowly drops (see Fig. 108).

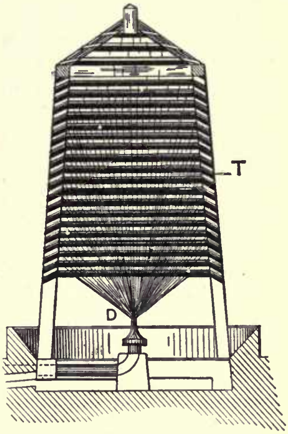

FIG. 108.

Finally we must mention the *direct union of the submerged condenser with the after-cooling apparatus in Linde's disc-cooler* (Figs. 109 and 110), which works exclusively with air propelled artificially. The condenser-worms lie in a rectangular iron box, *W*, in the cooling-water, which is kept in circulation by a stirrer, *R*, and in which a number of metal discs, *S*, are immersed to about one-third of their diameters; these discs rotate slowly about a horizontal axis, *X*, and so are kept continually wet. The layer of water on their surfaces then evaporates under the influence of a power air-current driven between the discs by means of the ventilator, *V*. In this way it is possible to evaporate from 1 to 2 lbs. of water per 10 sq. ft. of wetted surface per hour, corresponding with a heat-absorption of 1200–2400 B.T.U. Since, now, in the above-described condensers the evaporation takes place in the same way—although generally with a natural air-draught—the same numbers should also hold for them. On comparing these values with those experimentally obtained for the passage of heat through the heating surfaces of submerged condensers, it is seen that, *in general, atmospheric condensers require, roughly, double the surface of the former.*.

24. The Evaporators of cooling-machines serve for taking up the heat at a low temperature, so that the real cooling action

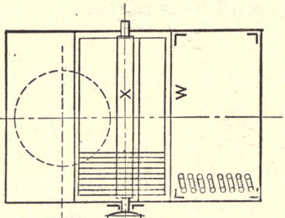

FIG. 110.

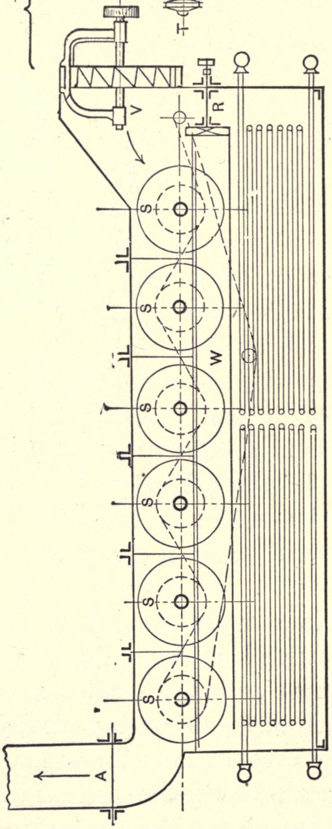

FIG. 109.

becomes completed in them. In most cases (in breweries always) the cooling action is transmitted to the objects or space to be cooled by a liquid which circulates between the latter and the evaporator, and which must neither freeze nor deposit solid matter at the temperatures obtained, as this would retard or entirely destroy its mobility. Such liquids present themselves in the solutions of certain cheap and easily obtainable salts, such as ordinary cooking salt ($NaCl$), calcium chloride ($CaCl_2$), and magnesium chloride ($MgCl_2$), while for temperatures above 0° ordinary water is sufficient (fresh-water cooling).

The condition that the salt solution must never separate solid matter means that it must not become saturated at any of the temperatures which it assumes in the course of the circulation, since separation of salt crystals would then begin on the slightest further lowering of temperature. These saturation quantities of the anhydrous salts, in pounds per gallon (10 lbs.) of water, are given in the following table[1] for various temperatures.

TABLE XV.

Temperature	$NaCl$	$CaCl_2$	$MgCl_2$
68° F.	3·60	7·4	5·70
50°	3·57	6·0	5·65
32°	3·55	5·0	5·60
23°	3·45	4·5	5·50
14°	3·35	4·2	5·40
5°	3·27	3·8	5·35
−4°	3·18	3·6	5·30

This table shows that the solubilities of the various salts are very different at the same temperature and vary with the temperature in different ways. Further, it is seen that saturation of the solution at ordinary temperatures (50° to 68° F.) is to be avoided, since separation would then inevitably occur on cooling. Thus, for example, if the salt solution attains a lowest temperature of 14° in working, its content of anhydrous $NaCl$, $CaCl_2$, and $MgCl_2$ should not exceed 3·35, 4·2, and 5·4 lbs., respectively, per gallon of water, even at ordinary temperatures. For the transmission of cold, then, only dilute solutions should be employed. But with these there arises the danger of freezing if their salt-content is too low. Information on this point is contained in the following table,[1] giving the freezing-points of solutions of NaCl and $CaCl_2$ of various strengths.

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[1] Derived graphically and partly by extrapolation from the values given in the physico-chemical tables of *Landolt* and *Börnstein*.

TABLE XVI.

Salt-content in Pounds per Gallon of Water.	Freezing-point for	
	NaCl	$CaCl_2$
0·5	25·2° F.	27·5° F.
1·0	18·7	22·0
1·2	16·0	19·5
1·5	12·2	14·8
2·0	6·0	5·4
2·5	0·2	−7·8
3·0	−4·7	

According to this table, a salt solution which is cooled to 14° should contain not less than 1·4 lbs. of anhydrous NaCl or 1·6 lbs. of anhydrous $CaCl_2$ per gallon of water. In practice, solutions containing 2·5 lbs. per gallon of water are generally employed, and these freeze at 0·2° (NaCl) or −7·8° ($CaCl_2$).

These salt solutions all have the disadvantage that they attack the iron of which the evaporator and the tubes mainly consist, while the best coatings offer no remedy. More recent investigations[2] have shown that this evil can only be removed by neutralisation, which is best done with soda (Na_2CO_3), from 1 to 2 lbs. of this being taken for 10 gallons of solution. Unfortunately this method is only applicable to common salt (NaCl), since the soda precipitates, from $CaCl_2$ and $MgCl_2$ solutions, the almost insoluble carbonates.

If the salt used for the brine or the water is not quite pure, or if the solution is prepared in dirty vessels, there is a danger of slime from the brine gradually depositing on the outer surfaces of the tubes and so injuring their heat-conducting power. This evil is most simply remedied by filtering the brine before introducing it into the evaporator. A suitable apparatus for this purpose, shown in Figs. 111 and 112, is manufactured by Messrs. B. *Weisser* of Bâle under the name "Satisfakteur." It consists of a metal chamber divided unequally by a transverse wall. In the larger division is a tank with perforated sieve-like walls for

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[1] From observations made by *Karsten, Gerlach,* and *Kohlrausch.*

[2] J. *Brand,* "Cooling Solutions which Do Not Attack Iron," Zeitschrift f. d. ges. Brauwesen, 1896. This author states that it is not necessary to add as much as the 5 per cent of soda which was originally made use of.

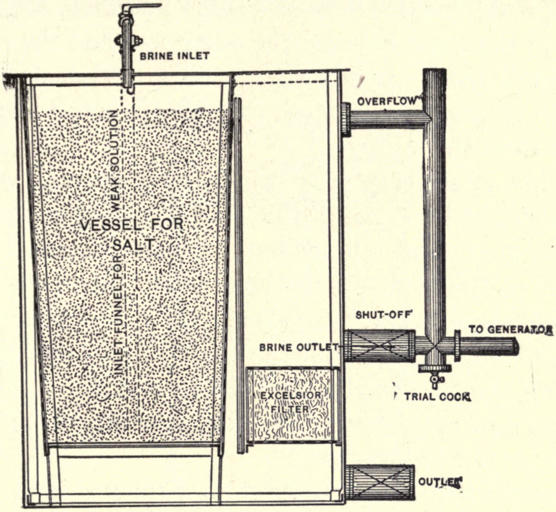


Fig. 111.

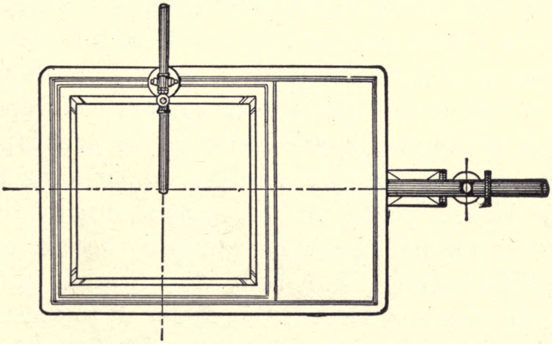


Fig. 112.

containing the salt to be dissolved, while in the smaller wood-fibre is packed between two sieve-plates. The weak brine or the water to be saturated can be either allowed to flow directly on to the salt, if the solution is to be quickly prepared, or gradually saturated in the space between the outer wall and the salt-tank of the larger division. In any case, the saturated solution, before leaving the apparatus, must pass through the wood-fibre, which retains all the solid matter.

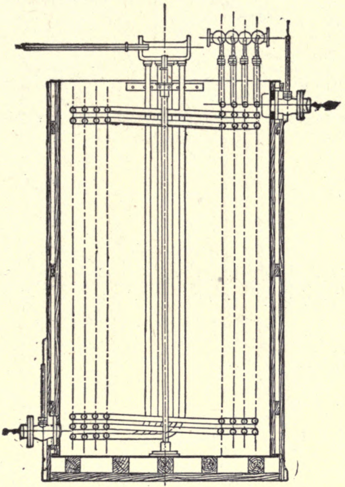
The *action of the evaporator* coincides, in general, with that of the steam-boiler, except that with cooling-machines the evaporation goes on at a lower temperature. The salt solution here serves as the heating substance, which surrounds the real evaporation space. The latter may, as in the steam-engine, have various shapes, according as the liquid introduced circulates during the evaporation or boils in a state of rest. The evaporators used are described as being according to the *circulation system* or the *sack system.*

The construction of the evaporators of the first kind, whether used for cooling salt solution or fresh water, corresponds generally with that of submerged condensers, as is seen from Figs. 113 and 114. They are mostly provided with a central stirrer. The circulation of the cooling-agent mainly proceeds so that the mixture of cold liquid with a certain amount of vapour which comes from the regulating-valve is carried, as far as is possible, to the lowest point of the coils, and the vapour formed is drawn off at the highest point; the brine, as shown by the arrows, enters at the top and, becoming gradually heavier as it cools, escapes at the bottom. It is here also advisable, in order that the separate coils may be made equal use of, to construct them of the same length, so that the distance between two coils is greater at the outside.

Characteristic for this type of evaporator is the increasing velocity in the tubes, which is caused by the continuous evaporation, and which favours the *carrying away of particles of liquid,* especially when the heating surface is only a scanty one. Circulation evaporators are hence always used, if wet vapours are to be drawn into the compressor, as is still always the rule with ammonia and carbon dioxide machines.

With a very large production of cold or very large fresh-water coolers, which, when used as reservoirs, often contain considerable quantities of water, there arises the difficulty that the number of evaporator coils and the diameter of the outer ones become largely increased, so that the action of the central stirrer on the latter is scarcely perceptible. In this case, several groups of coils, *S* (Figs. 115 and 116), are placed in a large vessel, which, it is

FIG. 113.



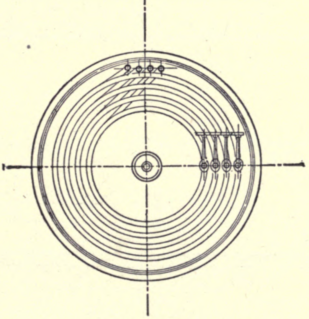


FIG. 114.

hardly necessary to state, must, under all circumstances, be well insulated.[1] Each group then contains its own stirrer, the pinions of which rest on carriers, *T*, and are driven by pulleys, *R*, one of the latter serving for several stirrers. In most cases the col-

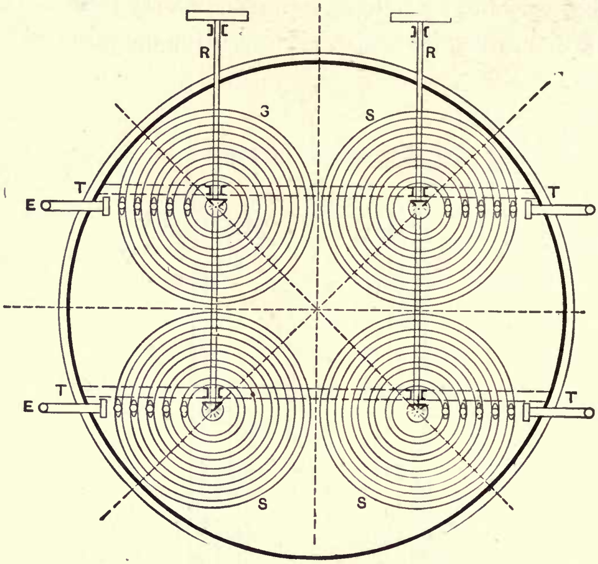


FIG. 115.

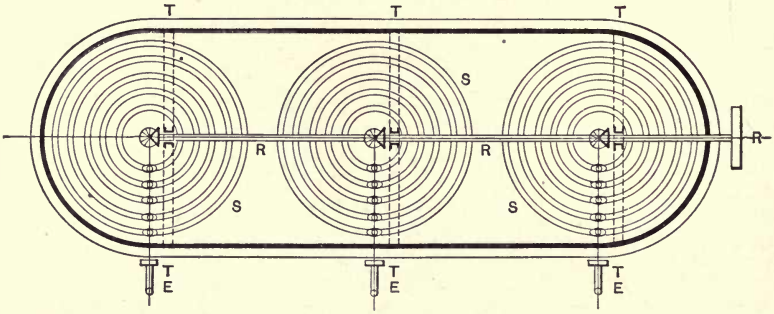


FIG. 116.

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[1] The insulation consists generally of a wooden jacket surrounding the wall of the vessel at a distance of 2·5–5 inches, the intervening space being loosely filled with cork chips, twig charcoal, peat dust, kieselguhr, etc.

lecting tubes, *E*, for the cooling-agent need not at first be joined up, as several compressors are required to draw off the vapour formed.

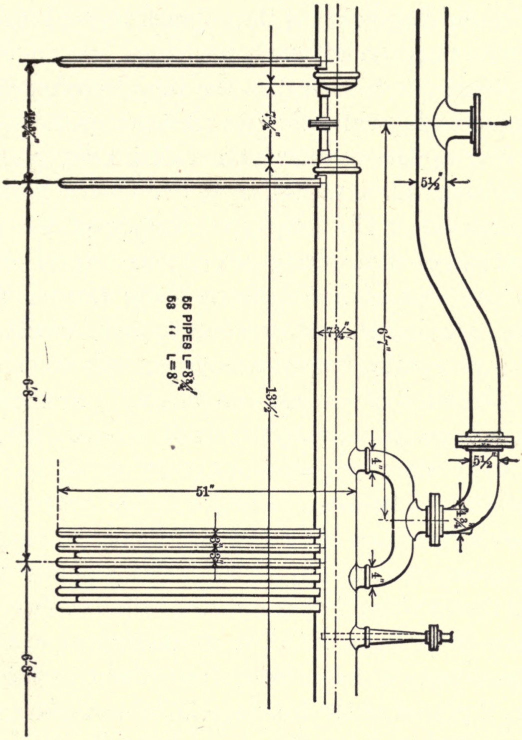

FIG. 117.

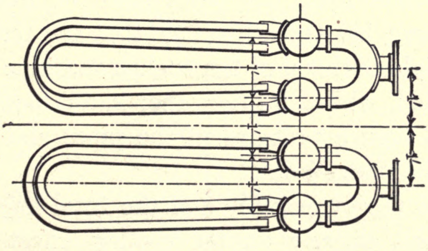

FIG. 118.

Figs. 117 and 118 show the construction of an *evaporator* recently made on the *sack system* for sulphur dioxide machines.

To two long and moderately wide tubes, which carry also the supports for the suction- and liquid-pipes, are fastened a large number of vertical U-tubes,[1] in which the evaporation of the sulphur dioxide entering at the top is carried on. Whether or not the tubes lying next to the supports are as active as those more remote must be left undecided.

In any case, it is seen that the liquid cooling-agent coming from the regulating-valve first gravitates into the U-tubes, afterwards giving up the vapour formed during the passage through the regulating-valve to the vapour-bubbles rising from the evaporator-tubes. The large cross-section of the whole of the tubes guarantees a moderate velocity for the vapour, and thus prevents particles of liquid from being carried along—which would be undesirable with sulphur dioxide machines—so that the vapour enters the suction-tube in a nearly dry state.

The circulation of the salt solution is, in most cases, effected by slowly acting centrifugal pumps with bronze stuffing-boxes and pistons, more rarely by piston-pumps, of which the cylinder and piston are then to be made of bronze. The velocity in the pipes should never be allowed to exceed 3 feet per second, since otherwise the resistances and the work of the pumps become very great and the cooling effect thereby impaired. The pumps themselves may either force into the circulation pipe, in which case the solution flows automatically from them to the evaporation vessel, or, if placed at the lowest point of the whole system, they draw the solution from the pipes into the evaporator. The first arrangement has the advantage that the pumps are in the immediate neighbourhood of the cooling-machine and are hence easily controlled, whilst, on the other hand, it gives rise to the danger of too high a pressure in the pipes and of consequent looseness of the flange-joints.

The size of the evaporator and the total length of its tube systems depend entirely on the cooling effect required; experience shows that, for a mean heating surface of 10 sq. ft., a heat transference of 3000–4000 B.T.U. may be calculated on. A higher value than this can only be attained by lowering the evaporator temperature and with this is combined a rise in temperature of the condenser, which also is overloaded, so that a considerable increase occurs in the work required.

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[1] From the drawings of the machinery installation of the Berlin Bock Brewery, erected by Messrs. *Schüchtermann & Kremer* of Dortmund.

The cooling effect may be conveniently determined in the case of evaporators working with brine circulation by measuring both the temperature change with thermometers in the entry and exit pipes (see Fig. 113) and, by means of a water-meter or other device (described in Chap. X), the quantity of salt water circulating per hour; the product of these two is to be multiplied by the *specific heat* (which for water is 1) of the brine. Since the measuring arrangements for the brine give results in volumes, it is necessary to ascertain the *specific gravity* by means of a hydrometer. Of these measurements, that of the specific heat is undoubtedly the most difficult on account of the manifold sources of error existing in such simple apparatus (so-called *calorimeters*) as is suitable to industrial uses. Consequently I have prepared the following small table, which contains the specific gravities and specific heats of solutions of salt, calcium chloride, and magnesium chloride. Owing to the impurities always present in the solutions and to the alterations of both magnitudes with the temperature, the results given here must be considered only as approximations.[1]

TABLE XVII.

Salt-content in Pounds per Pound of Solution.	Specific Gravity at 59–64°.			Specific Heat.		
	$NaCl$	$CaCl_2$	$MgCl_2$	$NaCl$	$CaCl_2$	$MgCl_2$
0	1	1	1	1	1	1
0·05	1·035	1·041	1·042	0·945	0·966	0·922
0·10	1·071	1·085	1·086	0·916	0·878	0·844
0·15	1·109	1·131	1·131	0·874	0·817	0·766
0·20	1·148	1·179	1·178	0·832	0·754	0·688
0·25	1·190	1·231	1·227	0·790	0·700	0·610

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[1] The numbers here given are taken from the well-known tables of *Lan-*

For instance, with a salt solution of specific gravity 1·1477, (spec. heat 0·832), which enters the evaporator at 28° and leaves it at 21° F., every pound of it gives up $0 \cdot 832 \times 7 = 5 \cdot 824$ B.T.U. If now 2000 gallons circulate per hour, the cooling effect of the evaporator will be $20{,}000 \times 1 \cdot 1477 \times 5 \cdot 824 =$ 133,700 B.T.U.[1]

If now the cooling effect cannot be obtained from the evaporator (owing to the lack of arrangements for measuring the circulating salt solution), the heat given up to the condenser may be determined; this must be diminished by the equivalent of the indicated compressor work (i.e., the measured horsepower multiplied by 2544). In all well-conducted installations such determinations should be made at least once a year in order to control the condition of the machine.

25. Distributing and Collecting Organs.—Both the condensers and evaporators of most cooling-machines consist, as we have seen, of a number of systems of metal tubes, which are joined at the ends by special parts in order to make connection with the main tubes leading to the compressor or to the regulating-valve. These parts are termed either *distributing* or *collecting organs*, according to whether they occur at the point where the cooling-agent enters or leaves the tubes.

Formerly these consisted merely of a cast- or wrought-iron body, with the interior of which, as shown in Figs. 119–122, both the separate tubulures and the main tube connect, regardless of their opposite positions and directions. A glance at these

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dolt and *Börnstein* in so far as they relate to the specific gravities and to the specific heats of sodium and calcium chloride solutions. The latter values for magnesium chloride are interpolated from researches of *Gumlich* and *Wiebe* (Zeitschr. für komprimierte und flüssige Gase, 1898).

[1] In the above example the losses by radiation and by the destruction of the stirring work are not included. If these did not exist, the temperature-difference and hence also the actual cooling effect would have greater values than those given above. These losses may be ascertained by observing the increase of temperature of the brine kept in motion in the evaporator by the stirrer, the compressor and pumps being stopped; in this way not only the quantity of salt solution but also the water value of the ice and the insulation are taken account of.

figures, which may be taken as typical for the cases where the main tube *H* enters at the side or in the middle, shows that in both cases the resistances to be overcome in passing from the main tube to the orifices numbered 1, 2, 3, 4, 5, or *vice versa,* are very unequal, so that, even when the resistances of the various tubes (made of equal length) have the same value, a regular distribution cannot be expected. This inequality becomes of greater importance the denser the flowing material and the greater the velocity with which it circulates in the machine. With cooling-machines at work, this unequal distribution among the separate systems caused by using distributors like those shown in Figs. 119–122, is shown principally at the evaporator by the different thicknesses of hoar-frost on the tubes close to the collector or distributor and is especially marked during the starting of the machine and shortly after stopping. In the first case, those tubes which offer the least resistance to motion (the inner ones, 2, 3, and 4 in Fig. 121) become more quickly coated, while the others remain black, and after stopping, the last-named tubes lose their covering more quickly. This of itself apparently unimportant matter has the drawback that

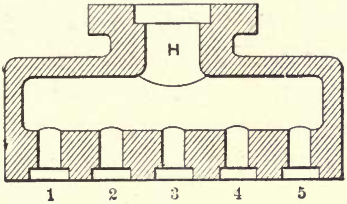


Fig. 119.

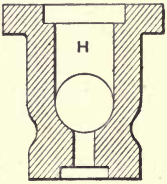


Fig. 120.

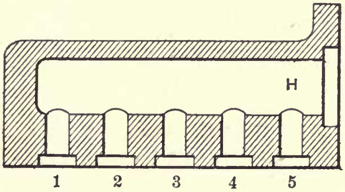


Fig. 121.

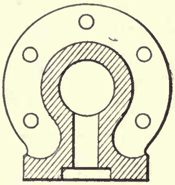


Fig. 122.

those of the tube-systems for which the total resistance is a minimum (even when all have the same cross-section) are preferred by the cooling-agent and are hence traversed by a larger quantity of the latter. From this it follows that quite different amounts of heat pass through the separate coils, so that the heating surface is only imperfectly utilised and therefore practically diminished.

The above-mentioned equality of length of the separate tubes is not alone sufficient to remedy this evil, so that *Linde's Ice Machine Co.*, which first used this arrangement, decided to combine the distributing chamber for the evaporator with the regulator and to provide this chamber with a rotating cock (see Fig. 40), which charges every tube with the same amount per unit of time.

Much simpler and yet equally satisfactory solutions of the problem may be arrived at by omitting the combination with the regulator, which should, even when it works automatically, be placed as conveniently as possible for the engineer in the engine-room—not too high or too low. Indeed the evaporator or ice-generator has frequently to be placed in another room on account of its size, and in this case it would be very inconvenient to combine the distributing vessel with the regulator. Further, improvement of the parts considered allows of the same model being used for the injection- and suction-mains of the evaporator and for the pressure-tube and liquid-tube of the condenser, which naturally means great simplification in the construction of the installation and in case of necessary interchanges. In the designing of uniformly acting distribution and collecting organs, the fundamental law to be followed is that *no one of the tubulures shall have a more favourable position or direction with reference to the main tube than any other.* If, for example, there are only two systems, a simple branched tube, i.e., a semicircular tube with the main tube at the vertex, fulfils all the conditions required in the collecting or distributing. A so-called T-tube would not be so good, owing to the opposite positions of the branches and to the sharp change in the direction. The semicircle, on the other hand, may be used also for a larger number of systems (Figs. 123 and 124) so long as the number is

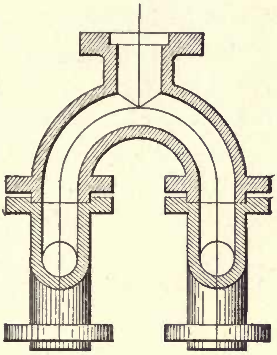


FIG. 123.

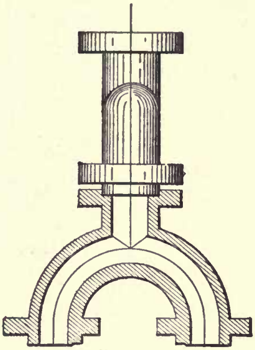


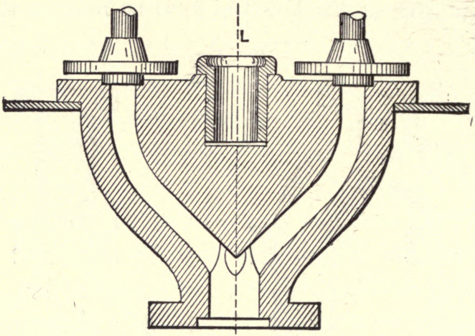
FIG. 124.

even. This limitation and also the increased number of joints —which is almost double as great as in the simple collectors (Figs. 119-122)—have prevented a widespread application of this most obvious construction.

If the arrangement of the tube-systems is such as to allow of the collecting organ having a central position, the simplest plan is to employ a *top-shaped branching,* which is further independent of the number of tubulures. Figs. 125 and 126 show such an arrangement for the lower collecting organ for condenser and evaporator, which at the same time closes the bottom of the apparatus and carries the bearing L of the stirrer. The tubes are bent at right angles just above their mouths and connected with the concentric spirals. This configuration has the advantage that in the erection and overhauling, all the tubes with the collecting piece can be lifted from the apparatus without loosening the flanges. Its application for the upper distributing piece would, however, scarcely be possible, unless the central stirring-apparatus were done away with. Equally unsuitable is it for the case when the tube-systems lie parallel to one another.

In such cases good results have been obtained by the use

FIG. 125.



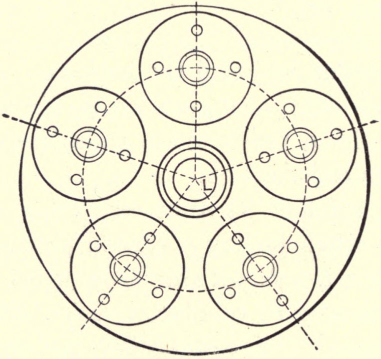
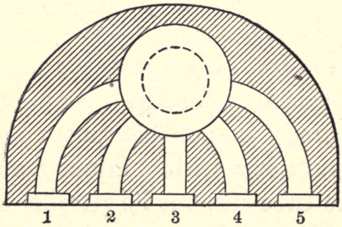


FIG. 126.



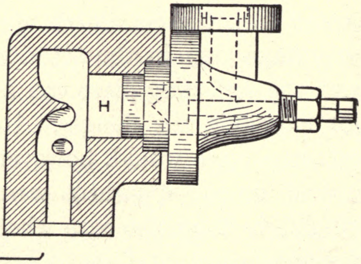


FIG. 127.

FIG. 128.

of a central chamber into which the main tube *H* (see Figs. 127 and 128) empties in an axial direction, while the separate tubulures go out radially from this. It is not necessary for the channels to be distributed regularly over the whole surface of the chamber, since they all cause the stream to deviate at right angles, no preference being thus given to any one of them. To the best of my knowledge, such collectors were first applied to cooling-machines by *Osenbrück*.

26. Mains and Their Connections.—The necessity of completely shutting off the cooling-agent in all its phases from the atmosphere requires, on the one hand, excellent material not only for the apparatus but also for the connecting tubes, and, on the other, special care in making the joints of the tubes. In all types of cooling-machine, *wrought-iron drawn tubes* are mostly used; occasionally, and only for sulphur dioxide machines, copper tubes are employed. The dimensions for the latter are to some extent arbitrary, but owing to the large number of wrought-iron tubes made, these are uniformly made in the following sizes.

TABLE XVIII.

Diameters of Tubes in Inches.	
Internal.	External.
⅜	⅞
½	1
¾	1¼
1	1½
1¼	1⅞
1½	2⅛
2	2¾
2½	3¼
3	3⅞

Of the numerous methods proposed and carried out for connecting tubes with one another and with other organs, we shall consider only the most important. The difficulties to be overcome consist partly in the fixing of the flange on the tube and partly in the tightening. These are quite got over by *Perkins's* method, which is largely used in England and consists in *pressing together two tubes, one with a sharp end and the other with a flat one,* by means of a union screwing on to the two tubes with opposite threads (Fig. 129). On opening such a joint it is found that the sharp end of one tube is pressed into the flat end of the other and slightly flattened, an excellent connection being thus made. The tightness of the screw is evidently not of importance. This only becomes necessary if the tubes do not come into contact, and it can be ensured by the use of leaden rings, which are pressed tightly into the threads of the screw either by the flange, as shown in Fig. 130 (*Case Refrigeration Co.*), or by small screws (see Fig. 131). In the latter case the tubes are first screwed together (*Tight-Joint*) and the lead then poured in and the screws inserted. The thread on the ends of the tubes is often made conical for the sake of security.

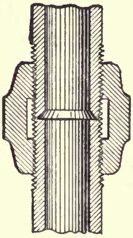


FIG. 129.

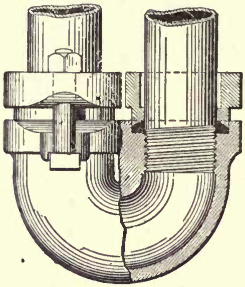


FIG. 130.

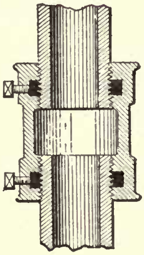


FIG. 131.

Flange tightenings are not required for the above-mentioned joints, which are seldom found in Germany. In using them care must be taken that they are not forced outwards by internal pressure—on which account smooth flanges cannot be employed—and that particles of the packing do not get inside of the tubes. With the American arrangement (*Frick Co.*) shown in Fig. 132, the latter is easily possible; this device has hence not met with approval in Europe. In Germany preference is given to the flange joints sketched in Figs. 133 and 134, in which the packing is completely embedded and can hence escape neither inwards nor outwards. This arrangement, in which the packing rests on a step, is cheaper to manufacture than the nut and screw device; the two methods are equally good in their working. The wrought-iron flanges themselves are well soldered, at their outer ends, with the tube and, occasionally, also at the inner ends. The thickness of the flange, and also the number of bolts, naturally depend on the pressure in the tube, which likewise determines the breadth of the tightening ring.

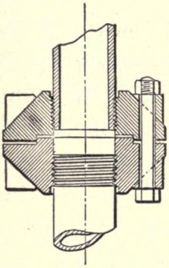


Fig. 132.

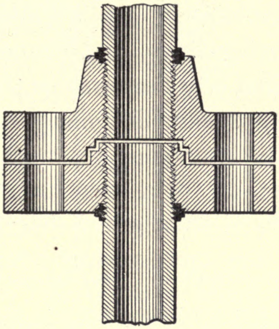


Fig. 133.

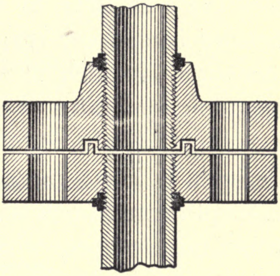


Fig. 134.

In general, the latter will not differ greatly from the thickness of the tube-wall. The tightening rings must, of course, not be made so broad that they protrude over the side-edges before pressing together, and further the screws must be cut as uniformly as possible.

CHAPTER V.

THE ERECTION OF ABSORPTION MACHINES.

27. Devices for Expelling the Ammonia.—As we have already seen (§§ 6 and 11), an ammonia absorption machine comprises, besides condenser, regulating-valve, and evaporator—the construction of which is essentially the same as that of the corresponding apparatus of compression machines—arrangements for expelling and absorbing the ammonia; the working of the latter depends in great measure on their construction. The earlier absorption machines suffered from grave defects, so that they could not compete with the more perfectly made compression machines. The machines made by the *Halle Co.* (formerly *Vaas & Littmann*) have, however, been used to some extent and are even to-day employed, though not so much as formerly. In describing absorption machines, we shall hence confine ourselves mainly to those of the above-named firm.

The expulsion of the ammonia is effected very generally by means of steam in so-called *generators*. The heating of this apparatus by direct fire, as is done in small installations, is unsuitable for large ones, as it does not allow of a sufficiently constant action. The generator consists of only two parts—a *horizontal cylindrical boiler* and a dome fixed thereon (Fig. 135). In the horizontal boiler, of which one end is riveted fast on while the other screws tightly in, lie a large number of *tubes* passing backwards and forwards and possessing no joints. The two ends of this heating coil are bent, one upwards and the other downwards, and pass, by means of screw supports with lead packing, through the wall of the generator into the open air, where the upper one connects immediately with a valve for shutting off or regulating the in-coming steam and the lower one with a separator for removing condensed water. In order to permit of the heating of the boiler and the expulsion of the ammonia being readily interrupted, the upper valve is also connected with a cold-water tube, which is naturally kept shut during the ordinary working.

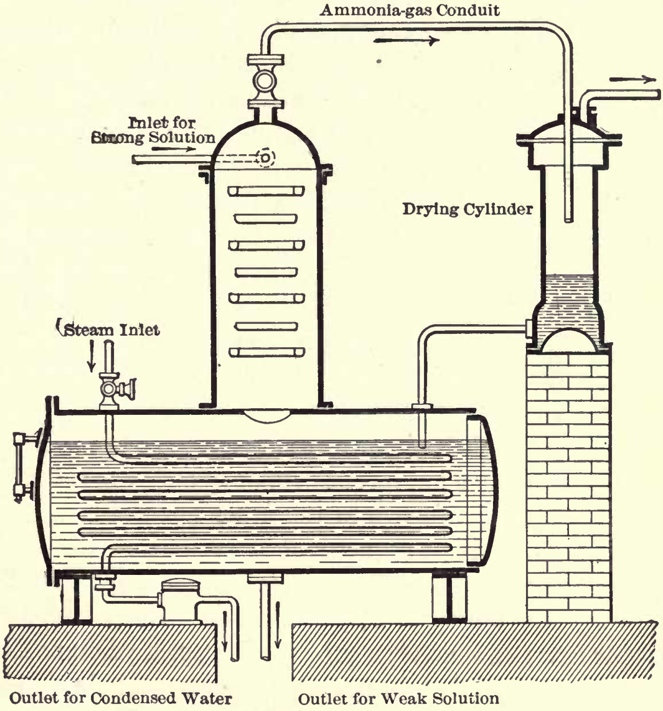


FIG. 135.

The coil in the generator is now completely surrounded by ammonia solution, the specific gravity of which, as we have already seen, diminishes as the proportion of ammonia present increases. In consequence of this the stronger solution lies in a layer above the weaker, which is drawn off by a tube at the lowest point of the boiler and flows through the interchanger to the absorber. The strong solution, on the other hand, is continually made up by fresh additions from above, this being effected by means of the central *dome* or *tower*, in which are arranged a number of *plates*, one above the other. Down over these plates runs the moderately warm strong solution coming from the interchanger and meets the hot ammonia-vapours rising from the boiler; at the same time a levelling of temperature takes place in the tower, while a little ammonia is also expelled from the concentrated solution trickling down.

Since, owing to the heating, a moderately high temperature prevails in the boiler and in the tower (also known as the rectifier), it is best to prevent the latter from radiating heat by a well-insulating jacket; the efficiency of this jacket determines, in a considerable degree, the amount of steam used by the machine. The whole apparatus is fixed on a foundation by two cast-iron feet sufficiently high to make the tubes underneath readily accessible.

With smaller machines heated directly, the whole apparatus is made in the form of a vertical boiler, the plates being then placed directly in the upper part. In any case, the boiler must be provided with a gauge-glass to show the height of the liquid in it, and also with a manometer.

The hot ammonia-vapours escaping from the top of the dome are accompanied by a not inconsiderable quantity of the solution, the introduction of which into the other apparatus might lead to irregularities. On this account a separator—the so-called *drying cylinder*—is placed immediately after the boiler in the gas-tube. This consists of a cylindrical cast-iron vessel, through the upper cover of which is a tube reaching some distance inside; the gases enter by this tube and escape at a point close to where the tube enters the cylinder. The gases are hence compelled to pass all around, so that the heavier liquid particles separate and are carried back by a tube from the bottom of the cylinder into the boiler. In order that the liquid may flow back of itself, the drying cylinder is placed, as shown in Fig. 135, on a tall pillar or a wall-bracket in the immediate neighbourhood of the boiler.

In his new absorption machines[1] *Habermann* gives a form quite different from the above to the generator, which is reduced to a series of horizontal, tube-like vessels, lying one over the other, the upper one carrying the rectifying tower (Fig. 137).

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[1] See *Belani*, "Absorption Cooling Machines," Zeitschrift d. Ver. d. Ingenieure, 1892.

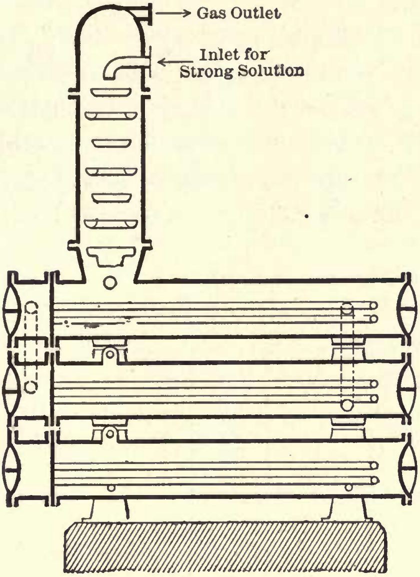

FIG. 136.

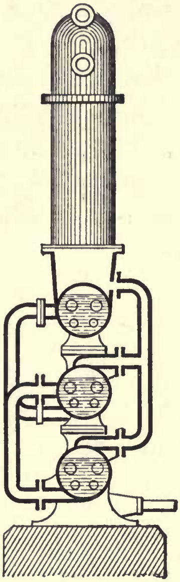

FIG. 137.

All the vessels are provided with supports at the top and sides; those at the top are connected with the common gas main leading to the tower, while the side ones serve as overflows for the solution which passes from top to bottom, becoming continually weaker. At one end of each vessel a double steam-chamber is screwed on, and from this several bent tubes jut out into the vessel; the fresh steam enters the lowest chamber first and escapes from the top one. The inventor has here worked on the right plan, namely, the prevention of the mixing of the strong and weak solutions or the facilitation of the separation. That the vapour should be given the opposite direction is, however, based on the erroneous view that the vapour, or condensed liquid, on leaving the boiler must have a considerably lower temperature than when it enters; when saturated vapours are used, this has no foundation. With this arrangement of the steam-heating, unless a water separator is provided for each vessel, difficulties may occur in the working, which are avoided if the steam passes in the opposite direction.

28. **Interchangers and Absorbers.** — Before the poor (i.e., largely gas-free) solution combines again with the ammonia from the evaporator, it is convenient to pass it through an apparatus in which it gives up the excess of heat it received in the boiler to the strong solution returning from the absorber. Such an *interchanger* (Fig. 138) is generally a cylindrical, air-tight vessel, which is traversed from the bottom *A* to the top *B* by the originally cool, strong solution, while the hot, weak solution passes through an inner spiral. In order to obtain a proper counter-current, as required for the most perfect taking up of the heat, the weak solution should be made to enter at the top at *C* and to escape at *D*.

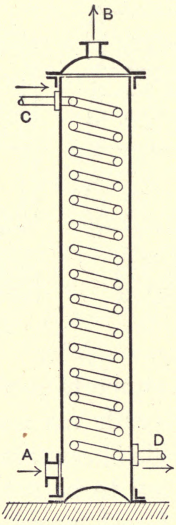

FIG. 138.

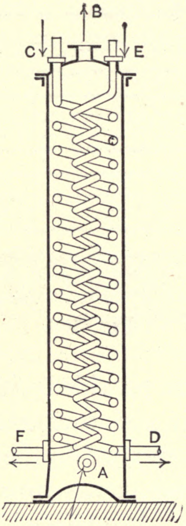

FIG. 139.

Better use is made of the heat taken up in the machine, and the condenser at the same time relieved, if the gaseous ammonia coming from the drying cylinder is passed through this apparatus. For this purpose a second spiral (Fig. 139) is introduced parallel to that used for the weak solution, and through this the ammonia passes from *E* to *F*.

In *Habermann's* absorption machines, to which attention has already been directed, these spirals for the gas and weak solution are separated, so that the strong solution first takes up the heat from the gas in one apparatus and then goes on to another, where it meets the weak liquid. Since, however, the temperatures of the weak solution and of the gas on leaving the boiler cannot be very different from each other, while the final temperature required is practically the same in the two cases, Vaas and Littmann's combination of the two processes appears more convenient than separating them according to Habermann; with the latter arrangement, the gas undergoes a greater cooling and the weak solution a less one, so that, although the work of the condenser is lightened, that of the absorber is rendered heavier.

The weak solution from this apparatus is sometimes carried through the condenser, or cooled still lower by cooling-water, before entering the absorber. This is done when the cooling-water is cold and abundant, because the strongly cooled solution is thus rendered capable of more readily absorbing considerable quantities of ammonia. This process goes on in the *absorber*, which Vaas and Littmann make in the form of an upright, cylindrical vessel traversed by vertical cold-water tubes. These tubes (Fig. 140) are fixed at the top and bottom into plates, and the two chambers thus formed are fitted with entry- and exit-tubes for the cold water, and also, in the case of the lower one, with a manhole for cleaning purposes. The cooling-water enters the lower chamber at *G*, takes up the heat of absorption, and rises gradually to the top, whence it flows away at *H*. The weak solution comes into the cylinder by the side-tube *J* at the top, while the ammonia from the evaporator is introduced at the bottom through *K*. In this way the solution becomes saturated as it sinks and reaches its greatest concentration at its lowest position, whence it is drawn as concentrated solution by the pump at *L* and carried through the interchanger to the boiler. The tube by which the ammonia is brought in at the bottom must at one part be higher than the level of liquid in the absorber, in order to prevent the aqueous solution from passing into the evaporator. The gauge-glass and the manometer of the absorber should, however, be carefully observed in those cases where an automatic float (which can be easily fitted to any absorber) is not employed to regulate the entry of liquid according to the level.

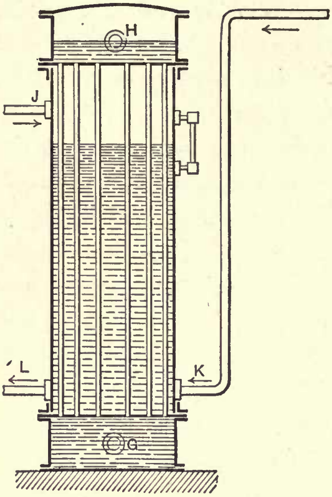

FIG. 140.

Quite different from this is Habermann's absorber, which is constructed just like his generator, described above. He introduces the weak solution at the top of the plate-tower, from which it falls into the different vessels, while the gas is led into all the vessels by a common tube. The tubes used in the generator for steam here carry cooling-water.

29. Working of Absorption Machines.—Besides the condenser and evaporator and the apparatus described above, the absorption machine comprises also a small pump, which has to increase the pressure of the concentrated solution from that of the absorber to that prevailing in the generator. This pump is generally vertical and single-acting and is driven by a shaft with fast and loose pulleys. If this is impracticable, it is best to use a direct-acting steam-pump. The attempt has often been made to replace this by an injector, in which the weak solution at the boiler pressure would draw the ammonia from the evaporator and compress it, but such an arrangement has not met with success. In this pump special attention should be paid to the stuffing-box, which is tightened with leather, woven cotton, and rubber rings.

In the erection and working of the machines, special care is to be taken that all the joints are tight. All the flanges must be turned and packed with rubber rings of the best quality capable of withstanding high temperatures. If all the parts, including the bent tube-joints, are distinctly marked, the erection of absorption machines can be carried out by any able fitter, so that it is unnecessary to employ an outside fitter. The machine is filled simply with concentrated ammonia solution, which is drawn in by the pump *S*. For this purpose a short tube is placed in front of the suction-valve, and to this a rubber tube dipping into the ammonia solution can be attached. When pumping in, all the cocks must be open except the one in the suction-pipe coming from the absorber. The pumping is continued until the gauge-glass in the heater indicates the normal height of liquid.

In starting the machine the cock in the weak-solution pipe must be closed in front of the absorber, in order to prevent premature entry of the concentrated solution into this apparatus; the heating is then gradually begun. In consequence of the expulsion of the ammonia, the interchanger slowly fills with the weak liquid and the condenser with liquid, anhydrous ammonia. The ammonia thus evolved displaces the air in the machine, and this should be allowed to pass from the drying cylinder through a tube and cock into water. When the formation of bubbles ceases in the water and the latter becomes heated, the air is all removed and the tube may be closed again. After a certain pressure is attained in the machine, the absorber may be filled by opening the tap in front of it; this draws so much liquid from the heater that the latter must be again filled by means of the pump. The pressure in the boiler is now raised to 8–10 atmospheres and the regulating-valve slowly opened, after which the evaporator-tubes—not at first covered with brine—become uniformly frosted. The regulating-valve is so adjusted that the manometer in the absorber shows an excess pressure of between 3·5 and 7 lbs. per sq. in., the cock admitting the weak solution to the absorber being regulated and the pump worked until the height of the liquid in the absorber vibrates regularly about a mean height. If now, when an abundant supply of cooling-water is passed through the absorber, the latter remains cold, the machine is in proper working order.

No smell of ammonia should be observable when the machine is working; care must be taken that the solution employed is not too weak. The weak solution taken from the heater during working should have a specific gravity of at most 0·918 (23° Baumé); if it has a greater value, some of it must be run away and strong ammonia solution added until the above number is reached. This test should be made once a week, when it should also be ascertained whether the machine still contains air. All the flanges should be frequently screwed up, especially when starting the machine.

In case of a stoppage or any disturbance in the working, the steam must first be shut off. Leakages in the machine may be readily detected by means of red litmus paper, which is turned blue by any escaping ammonia. When the working is interrupted for some time (as in winter), the pump-valves are taken out and greased and the generator thoroughly cleaned and provided with a new inner coating of red lead.

CHAPTER VI.

ON THE COOLING OF LIQUIDS AND KEEPING THEM COLD.

30. **Appliances for the Cooling of Liquids** are divided into two main groups: *evaporation apparatus*, in which the heat is taken up directly by the vaporisation of the cooling-agent, and *circulation apparatus*, in which the heat is taken up and carried away by means of a liquid previously cooled down in the evaporator of the cooling-machine. The devices of the first class correspond completely, in their action and construction, with the evaporators treated in detail in Chapter IV, so that here we need concern ourselves only with the circulation apparatus. The latter exhibit a certain complication compared with the evaporation apparatus, in that they introduce an intermediate substance and arrangements for effecting the circulation, but, owing to the greater available store of cooling-liquid, they possess far greater steadiness in working and are, to some extent, independent of the variations of temperature inside the evaporator. Where value is laid on these qualities, as is the case in breweries, such an apparatus is to be preferred to one of the evaporation type.

Circulation apparatus may be further divided into four different systems:

(*a*) So-called *boiler apparatus*, in which the liquid to be cooled down is at rest in a vessel, while the cooling-liquid either flows round this vessel or passes through it by means of pipes. In this way the temperature of the still liquid gradually falls to the required lowest point, when it is run off and its place taken by fresh liquid, so that the working is intermittent. Further, the lowest temperature thus obtainable will always be higher than that at which the cooling-liquid flows away.

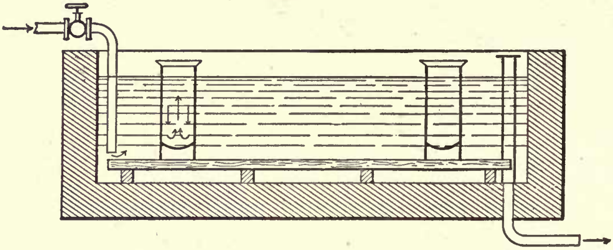

Fig. 141.

Such an apparatus is used more especially in dairies in order to cool either the milk for skimming or the centrifugated cream for butter, rapidly down from about 86° F. to 39–40° F. For this purpose cold fresh water in an open bath (see Figs. 141 and 142) is mostly used, while more recently a brine circulation apparatus constructed on the model of Field's double-tube steam-boiler (Fig. 143) is used, being dipped into the cream vessel for about a quarter of an hour. In all these apparatus circulation (as shown by the arrows) is readily produced in the apparently still liquid in consequence of the unequal cooling of its different parts, and this accelerates the cooling of the whole.

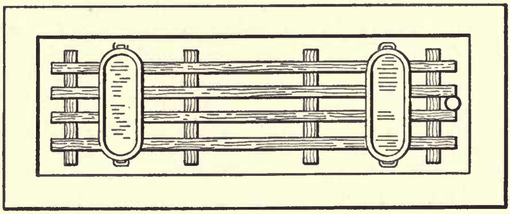

Fig. 142.

(*b*) *Parallel current apparatus,* in which one side of the surface through which the heat passes is traversed by the liquid and the other side by the cooling-liquid; here, also, the lowest temperature attainable by the body to be cooled down must lie somewhat above that at which the cooling-liquid escapes. On this account the more simple boiler apparatus is mostly preferred to this, and, in general, if the liquid to be cooled is to be kept moving, and it is also necessary to use the smallest possible surface, a

(*c*) *Counter-current apparatus* is employed. In this the liquids circulate in opposite directions on the two sides of the cooling surface, so that the entering cooling-liquid meets the escaping liquid to be cooled. In this way the latter may be brought down nearly to the temperature of the cooling-liquid. So that we have here the possibility of attaining, with a given cooling-agent, lower temperatures than can be reached in other forms of apparatus. If a pure counter-current action, such as we have already described in detail in the absorption of the liquid heat of the cooling-agent, cannot be effected, it is best to use a

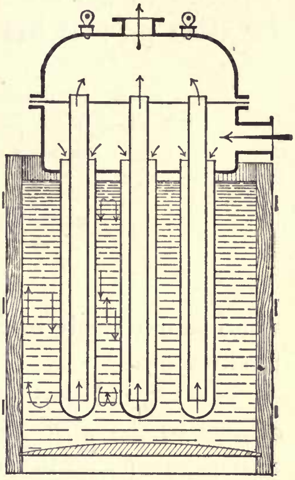

FIG. 143.

(*d*) *Combined apparatus,* which can, of course, never quite equal in its action the counter-current apparatus.

The simplest counter-current coolers are found in American breweries; they consist only of two copper tubes, one inside the other (see Fig. 144), the former being traversed by cold brine coming from the evaporator of the cooling-machine, while in the outer tube the hot wort from the copper circulates. The very considerable difference of temperature (about 120° at first) allows very small cooling surfaces to be employed, but leads to an enormous waste of cold. That this is the case is seen from the fact that the wort can be cooled from 150° to 55° F. by well-water, and only for a further reduction of temperature is it necessary to use ice-water or artificially cooled liquids. The American practice has a certain amount of justification only when an extremely rapid cooling is required for the purposes of the brewery.

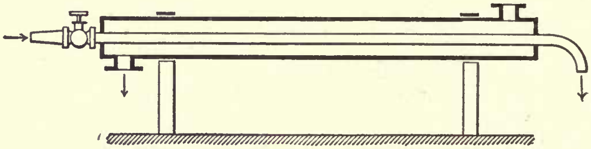

Fig. 144.

Further, it is not infrequently desired to allow of the access of air[1] to the wort (aëration) during the cooling process; this cannot be effected in such a closed cooler, but is readily obtained by allowing the wort to trickle over the outside of tubes through which the cooling-agent passes. If a series of thin-walled tubes are arranged one over the other in such a way that the cooling-liquid enters the lowest tube and flows through the others in an upward direction while the wort trickles down outside, each separate tube acts as a boiler apparatus, but the total effect is that of a very satisfactory counter-current action. The shapes of such wort-coolers (refrigerators)—the principle of which was laid down by an English engineer, W. *Lawrence*—are various, according to the cross-section of the tubes and the means of connecting them. The most widely used is the apparatus shown in Figs. 145 and 146, the tubes of which are nearly elliptical and made entirely of copper. The tubes support one another by means of a web and are arranged in two groups, those of the upper one being traversed by well-water and those of the lower by so-called ice-water, which is generally supplied by the fresh-water cooler of the brewery. The connection at the ends is made simply by means of vertical, box-like metal vessels, which are furnished with cross-walls and to which the entry and escape pipes of the cooling-agents are fitted. An advantage of this apparatus, which must not be undervalued, is the ease of cleaning, which is only required for the outside, as the interior is being continually washed by clean cold water.

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[1] This should then be sterilised, in order to avoid any infection of the wort.

The heat transference in such a thin-walled cooling apparatus (refrigerator) is found to be about 140 B.T.U. per sq. ft. per hour per 1° F. difference in temperature between the two sides of the cooling surface. If, for instance, 40 barrels (of 36 gallons) of wort have to be cooled from 160° F. to 40° F. in 2 hours without using a cooler, the cooling apparatus has to absorb, roughly, $40 \times 36 \times 10 \times (160-40) \div 2 = 864{,}000$ B.T.U. per hour.[1] With a rational mode of working, the cooling-plant is only required to bring the temperature down from about 57° to 40°, i.e., to absorb about 120,000 B.T.U. per hour, since the cooling from 160° to 57° can be readily effected by means of well-water, which enters at 50° and escapes at 95°. Thus, on one side of the cooling surface, a mean temperature of $\frac{1}{2}(160 \times 40) = 100°$ prevails and on

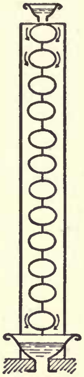

FIG. 145.

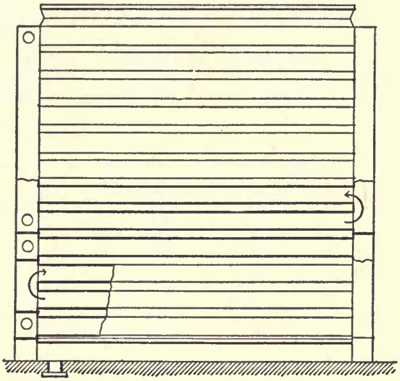

FIG. 146.

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[1] The specific heat may, with sufficient accuracy, be taken as 1. For the experimental numbers given above, I am indebted to a communication from the "Germania" Machine Co. of Chemnitz.

the other, one of $\frac{1}{2}(95+1)=48°$; the mean difference of temperature is hence $100-48=52°$, which, with a surface of 1 sq. ft., can remove $52\times140=7280$ B.T.U. per hour. The refrigerator must therefore have a surface area of approximately $864{,}000:7280=120$ sq. ft. In practice, indeed, it is usual to allow 6–7 sq. ft. of refrigerator surface per barrel of wort to be cooled per hour.

The "*Compagnie industrielle des procédés R. Pictet*" of Paris have designed a special form of counter-current cooling apparatus for the purpose of cooling down sodium sulphate solutions and subsequently separating the crystals in the cold.[1] The solution to be cooled enters the trough *BB* (Figs. 147 and 148) at 2 and escapes at 1, while the cooling-agent passes in the opposite direction inside a slowly rotating vessel. This rotating vessel consists of a hollow spindle, *FF*, on which hollow, lens-shaped bodies, *E* (see Fig. 149), are fixed by means of angle-irons, *e*. These hollow bodies are divided into two parts by partitions, *c*, which are carried on a rod, *gg*, inside the hollow spindle so as to make them stiffer, and are provided at the edge with a number of holes, *d* (Fig. 150); in each of these lens-shaped spaces the liquid flows outwards from the centre in the first half and in the

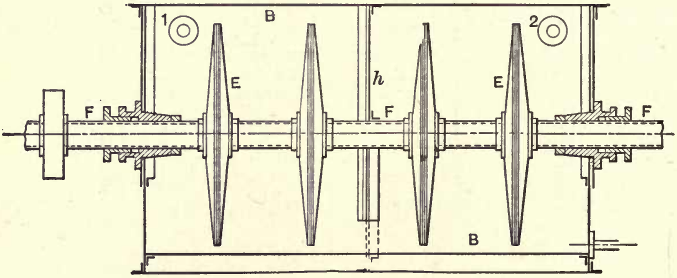

FIG. 147.

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[1] See Revue industrielle, 15, No. 19, and also Zeitschr. für Kälte-Industrie, 1895, page 41.

opposite direction in the second half, after which it passes through the hollow spindle to the next space. The entrance of the cooling-liquid to, and its escape from, the hollow spindle is effected by a bearing, *G* (Fig. 151), the interior of which communicates with the spindle by slits, *r*, in the latter, while the spindle is shut off from outside by a stuffing-box.

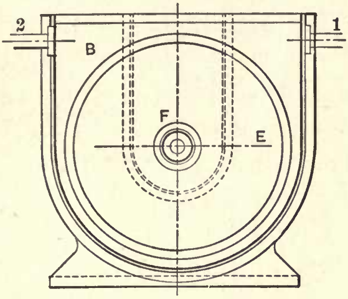

Fig. 148.

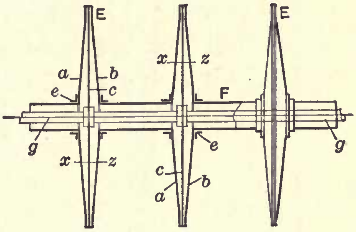

Fig. 149.

Practical data of this apparatus, which is used in South American saltpetre works, have not yet been published; in judging of its efficiency, an important place must be given to the work required to overcome the resistance to the motion of the cooling-agent, which has to make numerous sharp turns.

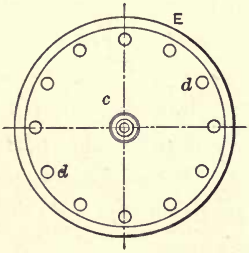

Fig. 150.

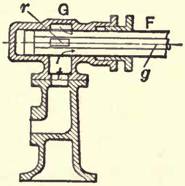

Fig. 151.

K. *Hirzel* of Winterthur, who has lately brought out a process for the partial separation of salt[1] from saturated brine by cooling it to 1° F., required a heat-exchange apparatus for the taking up the cold of the exhausted brine by the fresh saturated solution, and for this purpose he devised the appliance shown in Fig. 152. The warm brine enters at *A*, passes downwards through a large number of vertical tubes, *RR*, and is drawn cold from the apparatus at *B*, while the tubes are surrounded by the cold, exhausted brine entering at *C* and passing upwards between the partitions, *Z*, in a zigzag path. In this way the fresh 27 per cent salt solution from the borings is cooled from 50° or 54° to 10° F., while the cold brine which enters at *C* at 1° F. and contains only 15–17 per cent of salt escapes from the apparatus at *D* with a temperature of from 46° to 50°. The counter-current action is here a very perfect one, because the transference of heat, calculated from the dimensions of the apparatus, would be about 1100 B.T.U. per sq. ft. per hour for a difference of temperature of 9° between the two sides of the cooling surface. In the forms of apparatus described last, the outer wall must, of course, be well insulated from external heat.

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[1] See Zeitschrift für die gesammte Kälte-Industrie, 1896, page 141.

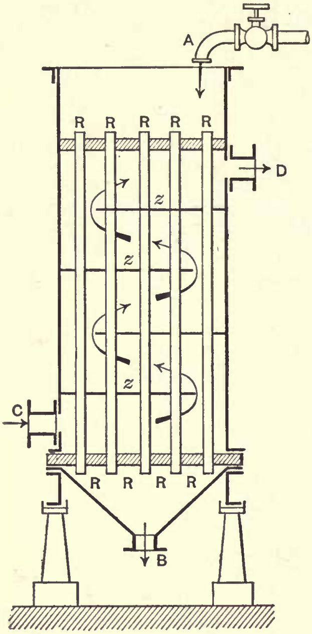

Fig. 152.

If the cooling of such salt solutions is carried still further, separation of crystals sets in. In Pictet's process the crystals form on the discs of a similarly constructed apparatus and are removed from these by scrapers and collected by a screw on the bottom of the vessel, from which they are removed by an elevator (chain-driven and provided with buckets). In Hirzel's process, however, the further cooling is effected by an evaporation apparatus, and the crystals sink to the bottom and as a pasty mass are transferred by a pump to a drying apparatus.

31. Apparatus for Keeping Liquids Cool are used mainly to absorb and carry away the heat set free in the interior of a liquid as the result of chemical processes, while the access of external heat to the liquid must be prevented by special air-cooling (see Chapter VII). Since the liquid has mostly to remain still for some special purpose (e.g., in fermenting wort, to allow the yeast to settle) and the temperature to be kept as uniform as possible

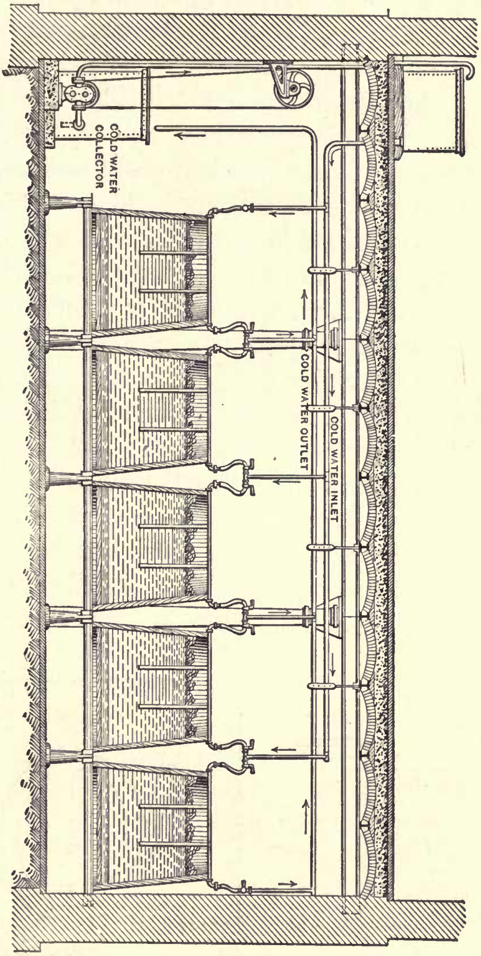

FIG. 153.

all through the liquid, movement of the latter through a tube apparatus is impracticable, while the withdrawal of the heat from the inside, as in vessels for cooking milk, is insufficient. It is, hence, necessary to effect the cooling in the interior of the liquid, in which the cooling-apparatus must be *immersed*. The construction of the apparatus is determined by the size of the vessel containing the liquid in which it is to be placed, as well as by the cooling surface required, and by the necessity of its being easily moved and cleaned. Such coolers are generally traversed by a stream of cooling-liquid, except when they are made in the form of floats and filled with ice. In most cases, the cooling-liquid employed is well-water cooled nearly to the freezing-point, and this flows down from the fresh-water cooler or a high cistern into a pipe arranged above the vessels and is led away by another pipe into a collecting vessel (see Fig. 153). These pipes are furnished with cocks and rubber tubes for connecting with the coolers. The warm water from the collecting vessel is pumped back from time to time into the cistern or the fresh-water cooler, where it begins a fresh cycle.

The coolers themselves are divided into two classes of *baffle-plate-coolers* and *tube-coolers*. They are always constructed of tinned copper plate and are connected by rubber tubes with the entry and escape pipes; the ends of the tubes rest on the edges of the vessel and are joined together by a rib, which imparts greater strength. The baffle-plate-cooler (Fig. 154) is a flat vessel divided up by horizontal partitions so as to form a zigzag passage

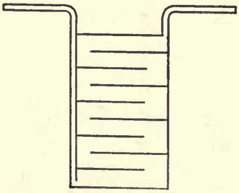

FIG. 154.

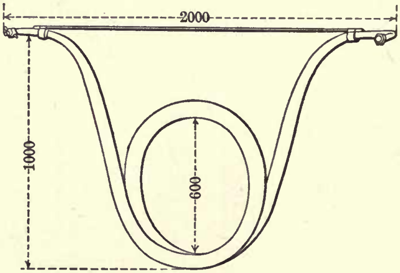

FIG. 155.

through which the cooling-agent ascends; it often consists simply of two plates soldered together and having the zigzag channel pressed between them. Tube-coolers are now almost entirely made in the form of a circular copper tube (as in Fig. 155) and are distinguished by their cheapness and by having no joints inside the liquid, but must have considerably larger dimensions than baffle-plate-coolers for the same cooling surface.

Since the liquid in the vessels must not undergo any change of temperature, the cold required is given entirely by the heat of the chemical reaction. Let us suppose, for instance, we have a wort with an extract of 25 lbs. per barrel (corresponding with 67·3 lbs. of dry extract), of which one-half is to be fermented; since the fermentation of 1 lb. of sugar liberates roughly 320 B.T.U.,[1] the heat to be withdrawn from each barrel of wort will be $320 \times 67 \cdot 3 \times 0 \cdot 5 = 10{,}768$ B.T.U. For a tun containing 40 barrels, the total cooling effect will be 430,720 B.T.U., or—distributing this over the whole period of fermentation, namely, 7 days or 168 hours—2560 B.T.U. per hour. If now the temperature is not to exceed 43° F., the cooling-water having an initial temperature of about 34° must escape from the separate coolers with a temperature not higher than 41°, so that at least $\frac{2560}{41-34}$ or 360 lbs. or 36 gallons of cooling-water must be supplied to each cooler per hour. The cooling surface of each cooler must since the mean temperature-difference between the liquid and, the cooling-water is 3½°, amount to 11–14 sq. ft.

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[1] According to *Buffard*, Comptes rendus, 1862, No. 8. See also Zeitschrift für Kälte-Industrie, 1895, page 215.

CHAPTER VII.

COOLING OF AIR.

32. Laws of Air-cooling.—The most important action of artificial cooling is the prevention or retardation of processes of decomposition, which, under normal temperatures, very soon commence in *food materials.* For these processes to begin, the access of air within certain limits of temperature is necessary, and also the presence of water, these being the conditions of life of the micro-organisms producing the decompositions.[1]

By lowering the temperature, which with solid bodies is brought about by the surrounding air, but can be effected directly in the case of liquids, the micro-organisms considered are not killed but are greatly retarded in their activity and, above all, their reproduction, especially if the cooling is accompanied by a more or less complete dehydration. The cooling industry of the present day strives, in the case of cooling installations connected with public slaughter-houses and market-halls, to *lower the relative moisture content of the air along with the temperature,* so that the absolute vapour-content must also fall. But this can only be attained by bringing the air to be cooled and dried into contact with surfaces at which a lower vapour pressure prevails than the saturation pressure of the water-vapour taken up by the air at the corresponding temperature. Cooling with cold water—at the surface of which there must always be a vapour pressure corresponding with the temperature—is, therefore, insufficient, even if it could be kept permanently below the freezing-point. On the other hand, in order to remove the undesirable excess of moisture, the air must be cooled considerably below the required temperature and only allowed to become warmer after the precipitated moisture has been removed. This is, however—quite apart from the extra work required by the lower cooling in the cooling-machine—attended with great difficulties, since such rise in temperature (by contact with warm surfaces, which must be practically anhydrous) is seldom to be effected without simultaneously raising the water-content, so that we are confined to cooling the air at surfaces having a vapour pressure lower than the saturation pressure.

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[1] That noxious parasites can be killed by long storage in cold- and freezing-chambers, has been shown by the comprehensive researches of *Ostertag* and *Reissmann.* See Zeitschr. für Fleisch- und Milch-hygiene, April, 1897.

Such surfaces are obtainable in two ways: by means of *dry ice* and by employing *salt solutions* which freeze with difficulty. In either case the action depends on the difference between the pressure of the vapour in the air to be dried and that at the surface of the body used at low temperatures. These differences of pressure are very small, but yet sufficient for the desired end, if large surfaces are used and the air is kept moving briskly. Thus, J. *Juhlin* (Swedish Academy of Sciences, 1892, Supplement) found the following pressures in inches of mercury:

Temperature.	Pressure.		
	Over Water.	Over Ice.	Difference.
8·6° F.	0·0687	0·0603	0·0084
14°	0·0865	0·0787	0·0078
23°	0·1261	0·1208	0·0053
32°	0·1818	0·1812	0·0006

For salt solutions these differences of pressure may be calculated from the molecular weight: from *Landolt* and *Börnstein's* Tables (Berlin, 1894) the following values are obtained for the differences between the vapour pressures of water and saturated salt solutions:

Temperature.	Calcium Chloride.	Sodium Chloride.	Magnesium Chloride.
14° F.	0·0450 inches	0·0081 inches	0·0082 inches
23° F.	0·0132 “	0·0180 “	0·0176 “

It is seen that for these solutions—which are most often used in practice, although not, as a rule, saturated—the difference in pressure is generally higher than for dry ice at the same temperature and the *drying action hence more intense.* The salt solutions become diluted by absorbing moisture from the air, so that their drying action gradually diminishes as time goes on; they must, therefore, be concentrated by running off a portion and either adding fresh salt to it or evaporating it. In the latter case it is best to insert a counter-current apparatus between the evaporating vessel and the air-cooler, so that the unavoidable loss of cold may be reduced to a minimum. The brine itself is either kept permanently at, or brought from time to time to, a low temperature by means of the evaporator of a cooling-machine.

In the cooling and drying of air at the *surface of dry ice,* which is placed on the tubes of the air-cooler (these being traversed either by the cooling medium of the machine or by a salt solution cooled by the latter), the moisture is precipitated in the form of ice, and the greater part of the micro-organisms of the air thus enclosed and rendered innocuous. On the other hand, when the cooling and drying is effected at the *free surface of salt solutions,* the moisture is only liquefied, so that a less (by the magnitude of the heat of fusion) quantity of heat has to be absorbed and removed for the same drying action. Since, further, when salt solutions are used, the surface may be increased to almost any extent in consequence of more intimate contact, this system seems to be the more economical.

In any case, there can be scarcely any permanent drying of air by contact with *wet ice* (natural ice), which is still often employed and before the development of the modern cooling-machine was exclusively in use. Both the cooling and also any precipitation of water from the air are here brought about only by the melting of a corresponding amount of ice, so that the surface remains always moist and hence cannot maintain a vapour pressure different from that of the moisture in the air.

Rapid removal of the melted ice does not alter matters, because the ice keeps moist unless it is subjected to immediate withdrawal of heat (as at the surface of the pipes through which the cooling-medium flows).

The *heat* required to be *abstracted* for the *simultaneous cooling and drying of the air* can be easily ascertained, if the initial and final temperatures and the corresponding moisture-contents of the air are known. The cooling alone of the air necessitates the withdrawal, for every pound of air, of a number of B.T.U. equal to the product of the difference of temperature and the specific heat at constant pressure, namely, 0·2377. Where the quantity of air is given by volume, i.e., in cubic feet, we must (since a cubic foot of air at 32° and atmospheric pressure weighs roughly 0 0868 lb.) multiply the difference of temperature by 0 0868 ×0·2377 =0·021.

So that, to cool, for example, 1000 cubic feet of air per hour from 36° to 27°, 1000×9×0 021 =189 B.T.U. would have to be absorbed per hour, without taking into account the moisture-content.

The heat to be *withdrawn* on account of the *drying of the air* may be obtained by means of the following *saturation table,* giving the water-vapour content of saturated air (in lbs. per cubic foot) at various temperatures.

TABLE XIX.

Temperature, Degrees F.	Moisture in Pounds per 1000 Cub. Ft.	Temperature, Degrees F.	Moisture in Pounds per 1000 Cub. Ft.	Temperature, Degrees F.	Moisture in Pounds per 1000 Cub. Ft.
−4	0·071	24	0·224	38	0·379
+5	0·104	26	0·242	40	0·408
14	0·150	28	0·261	42	0·440
16	0·164	30	0·286	44	0·472
18	0·177	32	0·304	46	0·505
20	0·192	34	0·330	48	0·540
22	0·207	36	0·351	50	0·584

The heat set free by the precipitation of 1 lb. of water from the air within the range of temperature covered by this table may, with sufficient accuracy, be taken as 1100 B.T.U. To calculate the total heat liberated in this way, the moisture present in 1000 cub. ft. of the air at the final temperature is subtracted from that present in the unsaturated air at the initial temperature and the difference multiplied by 1100 and by the number of thousands of cubic feet of air.[1]

Let us suppose we have, as above, 1000 cubic feet of air, which at 36° contains about 90 per cent of the quantity of water-vapour required for saturation, and which is to be dried by cooling to 26°; the heat to be withdrawn owing to the precipitation of moisture is: $1100(0 \cdot 351 \times 0 \cdot 9 - 0 \cdot 242) = 81$ B.T.U., while the lowering of temperature of the air requires the removal of 189 B.T.U. (see above). This example indicates the great *importance of taking into account the cold used up in the air-drying,* if the danger of a considerable under-estimate is to be avoided.

The best means of ascertaining the moisture-content of the air is by the use of the *hygrometer,* which consists of two thermometers indicating tenths of a degree, the bulb of one being kept continually wet by a moist flap. By the evaporation taking place at this bulb—which increases in intensity with the dryness of the air—heat is absorbed, so that the wet thermometer will show a lower temperature than the other. The difference between the two temperatures (the so-called *hygrometric difference*) gives a measure of the relative moistness of the air, the values of which are contained in very detailed tables. On the opposite page are given the values for the temperatures with which we have to deal, the barometric pressure being 29 72 inches.

If, for example, the dry thermometer indicates 34° and the wet one 38°, the hygrometric difference is 4°, and for this the table gives a moisture-content of 0·63. Since 1000 cubic feet of air saturated with moisture at 38° contain (see Table XIX) 0·379 lb. of water, they will contain, with a moisture-content of 0·63, only $0 \cdot 63 \times 0 \cdot 379 = 0 \cdot 239$ lb. and can, therefore, take up $0 \cdot 379 - 0 \cdot 239 = 0$ 140 lb.

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[1] It should be pointed out that the above method of calculation is an approximate one, which, however, gives sufficiently accurate results for the limits of temperature of cooling-machines; it is to be preferred, on account of its great simplicity, to the exact determination.

TABLE XX.

Temperature by Dry-bulb Thermometer.	Difference between Dry-bulb and Wet-bulb Readings.			
	2° F.	4° F.	6° F.	8° F.
14° F.	0·64			
16	0·66			
18	0·68	0·42		
20	0·70	0·46		
22	0·71	0·49		
24	0·73	0·52		
26	0·74	0·55	0·40	
28	0·75	0·57	0·43	
30	0·77	0·60	0·45	
32	0·78	0·61	0·48	0·37
34	0·79	0·63	0·50	0·40
36	0·82	0·66	0·53	0·42
38	0·83	0·68	0·56	0·45
40	0·84	0·70	0·58	0·47
42	0·84	0·71	0·59	0·49
44	0·85	0·72	0·60	0·50
46	0·86	0·73	0·61	0·51
48	0·86	0·73	0·62	0·52
50	0·86	0·74	0·63	0·53

It may be mentioned that the *hair hygrometer*, which is largely used, is very unreliable and should always be adjusted before use. The wet- and dry-bulb hygrometer is greatly to be preferred to all other types.

Air-cooling apparatus used in practice in conjunction with cooling-machines are of two kinds: (1) Those in which the air impinges on cold tubes on the outside of which the moisture of the air is almost always precipitated as a thin film of ice; these are known as *tube air-cooling apparatus.* (2) Those in which the air is brought into contact with the surface of the cooled salt solution itself, such being termed *brine air-cooling apparatus.*

33. **Tubular Air-cooling Apparatus**, which we shall first describe, may be arranged either for *direct evaporation*, in which case the evaporating cooling-agent circulates in the tubes and no intermediate liquid is used; or for *indirect evaporation*, where a salt solution, cooled in the evaporator of the cooling-machine, traverses the tubes. Owing to the lack both of a sufficient theoretical basis and of reliable comparative experiments, it is not at

present known which of these two methods is practically the better. At first sight it would appear as if the direct evaporation method—which is in great favour in America and in Europe is adopted by Messrs. *Fixary* of Paris, by the *Humboldt* Machine Company of Kalk-Köln, and by Messrs. L. *Seyboth* of Munich—would give the best results, as, in consequence of the omission of the intermediate liquid and hence of at least one heat-transmission surface, the difference of temperature between the air and the cooling-agent circulating in the machine would become smaller and the work used by the cooling-machine therefore diminished. But (just as with surface condensers), owing to the relatively lower transmission of heat through surfaces on one side of which evaporation occurs, this is only possible if the surface, and hence also the total length, of the tubes is made very great, and if this is done, the resistance to the motion of the cooling-agent is increased. Further, direct evaporation of the cooling-agent in the tubes is found to require very careful regulation of the cooling-machine, and, unless a large quantity of brine is interposed to act as a reservoir for the cold, deviations of temperature of the machine are easily transmitted to the air and so disturb the stability of the conditions. On account of the great importance attached to this point in Europe, direct evaporation has not been very widely used here, although, as the author has shown with the installation in use at the public slaughter-house in Munich, it allows the moisture-content of the air to be kept very low in consequence of the low temperature of the ice adhering to the evaporator-tubes.

Of more significance than the above classification of tubular air-coolers is that which differentiates them according to whether the *air circulation* is *natural* or *artificial.*[1]

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[1] Tube-cooling with natural circulation of air is, in the case of direct evaporation, to be regarded as the inverse of so-called steam-heating, and, when indirect evaporation (brine circulation) is employed, as the inverse of *heating by hot water.* Cooling with artificial circulation of the air corresponds with the so-called *air-heating,* in which, on account of the considerable difference in temperature between the air leaving and that entering the heating apparatus, no ventilators are necessary.

(*a*) *Cooling apparatus with natural air circulation* are placed, in the form of a bundle of pipes, either in the *cold-room* itself or *above the latter in a special chamber* connected with the room by channels. Since the air in the cold-room (e.g., a meat store, the fermenting- or lager-cellar of a brewery, etc.) becomes warm, it assumes an upward motion, whilst that in the neighbourhood of the cold pipes is cooled and dried by these and falls by virtue of its greater specific gravity.

In Europe it is now usual to hang the bundle of cooling-tubes from the ceiling of the cold-room (Fig. 156), by which means a descending current of air is produced in the middle of the room and an ascending one at the sides. In America, on the other hand, the tubes are generally fastened horizontally to the side walls, the current of air then flowing in the opposite direction (Fig. 157). To increase the cooling surface, the tubes have of late years been provided with ribs, either cast with the tubes or fixed on afterwards.

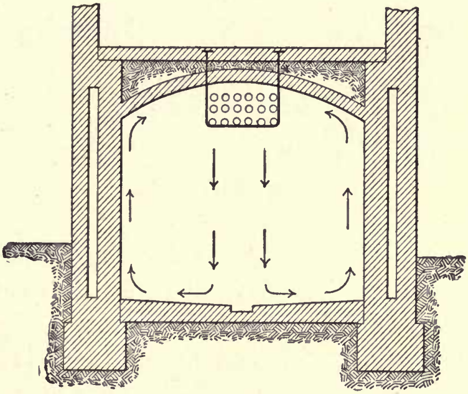

Fig. 156.

The American plan has the disadvantage that the coldest air from the tubes comes into contact with the walls and afterwards with the floor, the surfaces of which are mainly the ones through which heat penetrates from outside. With the European method, the air is only brought into contact with these surfaces after it has taken up at least the greater part of the heat in the interior, so that the difference of temperature between the walls and floor, on the one hand, and the external air, on the other, is diminished, as also, therefore, is the penetration of heat. Against this must be set the fact that the American arrangement favours the drying of the floor and walls, which the dry cold air first passes over, whilst, when the pipes are hung from the ceiling, the air impinges on the floor and walls only after it has become warmed and saturated with moisture in the room. In the cooling installations of breweries this last circumstance is of no great moment, so that the European method is the more economical in their case; but for preserving food the side arrangement of the tubes is the more suitable, if it is not preferred to make use of artificial circulation of the air (see below).

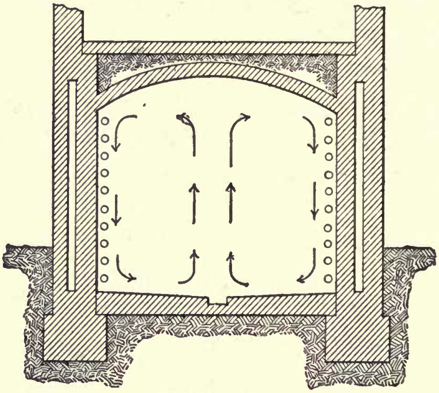

FIG. 157.

No matter which of the above methods is employed, care must naturally be taken to immediately catch and syphon away the thaw-water dropping from the tubes when the machine stops, as it contains a large number of noxious germs. This is most conveniently effected when the cooling-apparatus is arranged in a separate upper chamber (Fig. 158), where the tubes are simply placed over a drip-pan, which does not interfere with the circulation. This upper chamber, which should be well insulated like the cold-room itself, is made of such a size that a man can easily move about in it and is connected with the room by side channels up which the warm air passes, and by one or more channels, most suitably central ones, by which the cold, dry air proceeds back to the cold-room. In order to prevent this cold air from mixing too soon with the air of the cold-room, the cold-air passage is made in the form of a tube, open at the bottom (see Fig. 158).

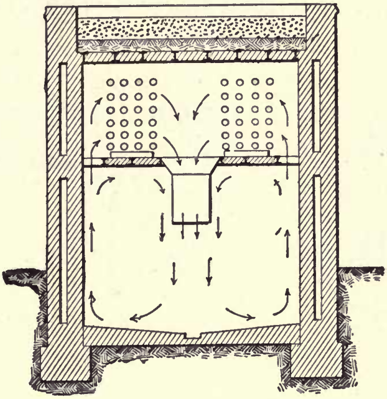
FIG. 158.

Characteristic of all the above-described tubular air-coolers with natural air circulation is the slowness of movement of the air, consequent on the small differences of temperature coming into play. This is always advantageous when it is mainly a question of keeping a room cool, but *not when an energetic drying* of the stored goods is aimed at, as this can only be attained by active circulation and frequent renewal of the air.

So that cooling with natural circulation of the air answers perfectly to the requirements of *breweries,* in which any attempt to dry or take up moisture from the large quantities of liquid in the fermenting- and lager-cellars would be completely frustrated. Of quite another nature are the conditions in *cooling installations for readily putrescible food materials* (meat, fish), for preserving which at least a surface drying is required.

The cooling-tubes used for brine circulation are mostly welded flanged tubes of 2 inches internal and $2\frac{1}{4}$ inches external diameter,[1] their mean heat-transmission surface being hence 0·528 sq. ft. per foot-length (Figs. 159 and 160). In consequence of the very great retardation of the heat-transmission by the frost on the tubes, the withdrawal of more than about 35 B.T.U. per hour per sq. ft. should not be reckoned on, and when direct evaporation is used it is best not to exceed 30 B.T.U., as otherwise the internal temperature has to be too low. The working is most favourable when the frosting on the tubes is only heavy enough to appear gray; tubes thick with ice indicate a large waste of work. The intensity of the frosting depends mainly on the initial moisture-content of the space to be cooled; a stable condition is only reached when no more ice is deposited and no thawing takes place. If the temperature of the air in the room is then above the freezing-point, that of the surface of the ice-layer is exactly 32°, independently of its thickness and of the temperature inside the tubes.

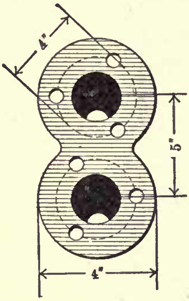

FIG. 159.

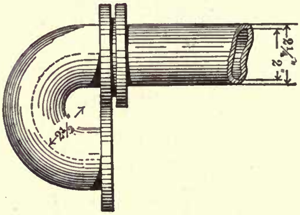

FIG. 160.

(b) *Tube apparatus with artificial air circulation* must be arranged in separate chambers in order to bring the air into contact with the surfaces of the tubes. By means of two channels these chambers are then placed in communication with the tubes by which the cold air is introduced to, and escapes from, the cold-room. In the collecting channel of the escape-pipe is placed the ventilator, which requires the less work[1] for driving it the greater the section of the air-mains and the less the number of sharp bends or angles. If possible, this apparatus also should be placed above the cold-room so as to utilise the velocity conditioned by the difference in temperature between the ascending and descending air; if the cooling-apparatus were placed beneath the cold-room, this natural movement of the air would have to be annulled by the ventilator. This is all the more noteworthy because the work used in operating the ventilator is entirely converted into heat and so diminishes the cooling effect by the amount of its heat-equivalent. With a clumsy arrangement the loss thus incurred may be very considerable.

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[1] The normal length of these tubes is 5 yards and the weight of one tube with its two flanges about 42 lbs.; each connection (see Figs. 159 and 160) weighs about 8·5 lbs.

The *ventilators* generally used are axial ventilators and screw-blowers, these being preferred to centrifugal ventilators, which, in consequence of the narrowing of the air-passage and the frequent change of direction of the stream, absorb much more work than those of the first two classes; further, centrifugal blowers favour absorption of heat from outside, owing to their large external metal surfaces.

When a ventilator is used, the *circulation of air inside the room* is completely under control by the arrangement of the entry- and escape-channels. The latter, made formerly of galvanised iron[2] but now almost exclusively of wood previously carefully impregnated with iron vitriol,[3] are mostly rectangular tubes and are fixed to the ceiling of the cold-room. The apertures through which the air enters the room are on the lower side of the entry-channel, and those by which the air escapes,

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[1] Under average conditions, with an air velocity of 4 to 5 yards per second in the main channels, we may reckon that the work to drive the ventilator amounts to 0·15–0·25 horse-power per 1000 cubic yards of air delivered, and by this a rise of temperature of 0·5–0·9° F. is produced. By doubling the quantity of air and hence the velocity, the work used and the rise of temperature would be quadrupled.

[2] These have the disadvantage that water is readily deposited from the air on the inside, whence it drops and contaminates the goods and nullifies the drying

[3] A large number of new methods for impregnating wood have recently been investigated. Particulars of these are given in the Zeitschrift für Kälte-Industrie, 1897 and 1898.

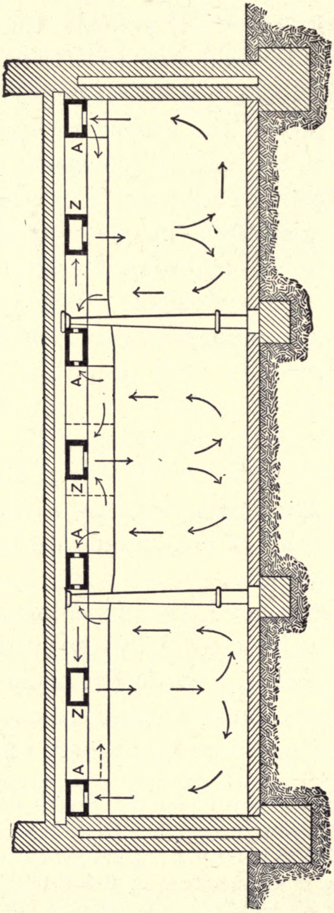

Fig. 161.

in the side faces or, more rarely, on the upper or lower side of the exit-channel. The motion of the air taking place in the room is then that indicated in Fig. 161 by the arrows. The arrangement of the channels is shown in plan in Figs. 162 and 163. It has also been found convenient to place the exit-channels above the pathways, while the entry-passages deliver the cold air immediately above the compartments filled with goods

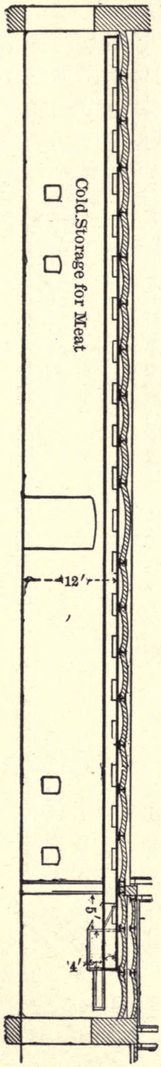

Fig. 163.

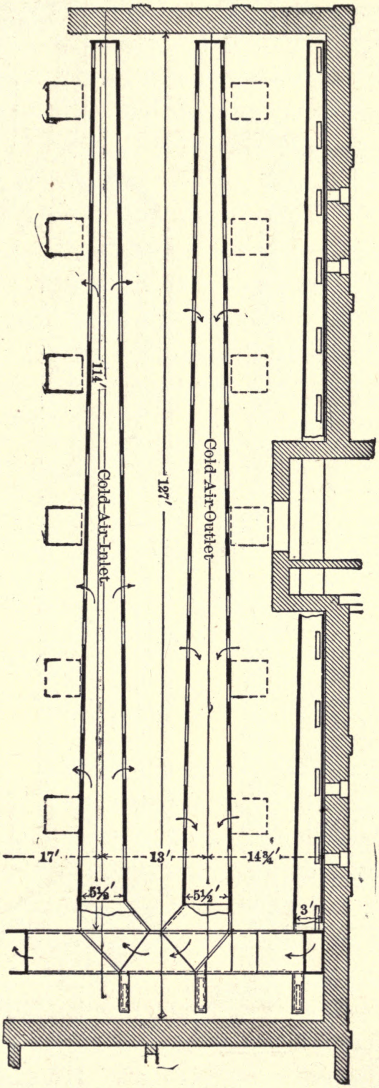

Fig. 162.

these compartments being shut off from one another and from the paths by means of sliding gratings.

As an example of a tube-cooler which has given good results practically, and which works with direct evaporation, we show in

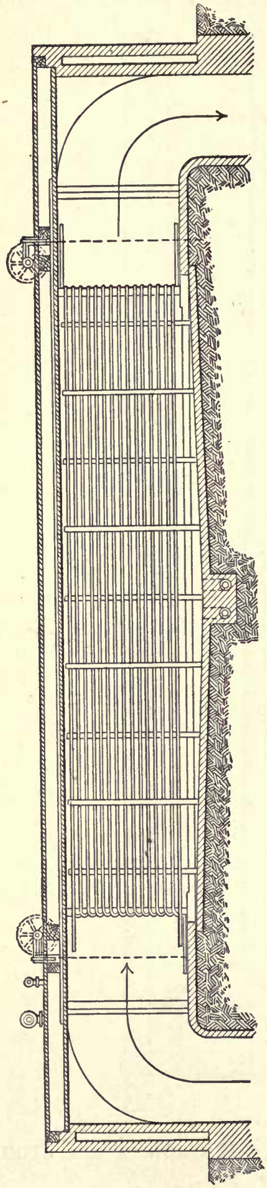

Fig. 164.

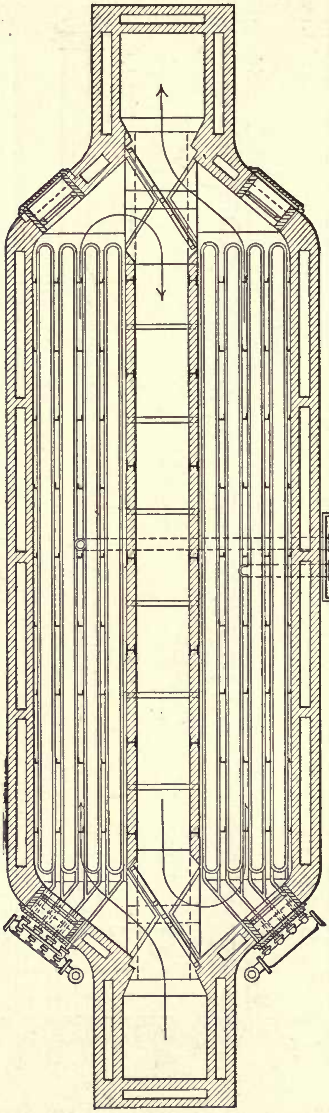

Fig. 165.

Figs. 164–166 that constructed by the *Humboldt* Machine Co. It consists of two tube-chambers separated by a passage, only one set of tubes being always traversed by the evaporating cooling-agent (ammonia in this case). The air coming from the cold-room first thaws the frost on the tubes in the other chamber owing to its higher temperature, and then traverses the separating passage and finally the second tube-chamber, where it becomes more intensely cooled and dried. When, after some hours, the tubes in this latter chamber become frosted, the ammonia is shut off from them and led into the tubes of the other chamber; the air-current is also changed over by the valves shown in Fig. 165. The movement of the valves is effected by means of hand-wheels attached to screw spindles on the cover of the apparatus. Both are shown in Figs. 164 and 166.

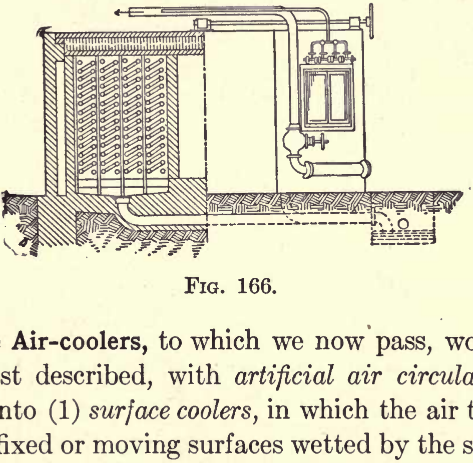

FIG. 166.

34. **Brine Air-coolers,** to which we now pass, work, like the apparatus last described, with *artificial air circulation.* They are divided into (1) *surface coolers,* in which the air to be cooled impinges on fixed or moving surfaces wetted by the salt solution, and (2) *spray-coolers,* in which the air is exposed to the action of the cooled salt solution in the form of a more or less finely divided rain.

Surface coolers with fixed surfaces do not differ in their arrangement from the so-called evaporator condensers of steam-engines and cooling-machines (see § 23). They consist of horizontal tubes which run backwards and forwards, one above another, so as to form a vertical grid; inside the tubes the cooling-agent evaporates, while down the outside trickles the brine, which takes up heat and moisture from the air impinging on it. The whole is placed in a chamber connected with the entry- and exit-channels for the air. The trickling brine is collected in a basin from which it is pumped back into the distributing trough above the tube-system.

By this arrangement, spurting of the brine into the air cannot very well be avoided, so that lately it has become more usual to allow the cooled brine to flow down either vertical corrugated plates (Fig. 167) or flat plates slightly inclined to the horizontal (Fig. 168), the air being led between these plates as nearly as possible as a counter-current. In this case the brine is collected in a trough at the bottom of the plates and is carried off sideways, out of the air-current, into the basin under the air-cooler.

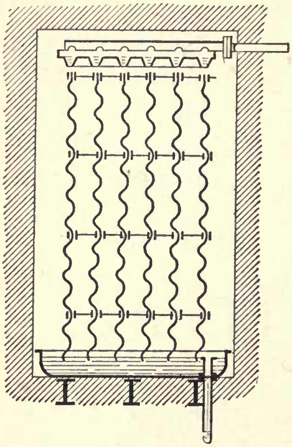

Fig. 167.

Movable surface coolers have become widely used in the form of Linde's *disc coolers* (Fig. 169). In these the brine is placed in a trough, at the bottom of which the evaporator tubes lie, while in the upper part are a large number of metal discs from 4 to 5 feet in diameter, carried in groups of 50 to 80 on rotatable horizontal axes. By means of pulleys a certain number or all

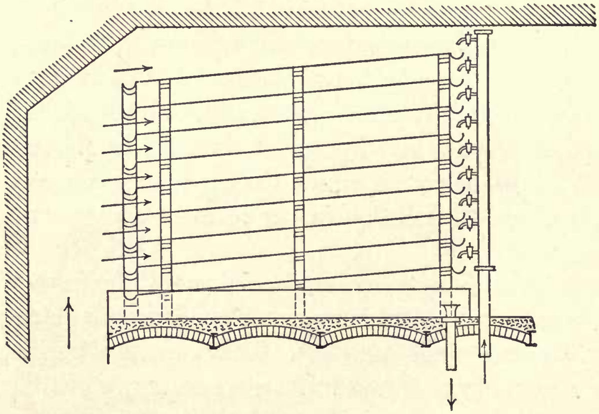

Fig. 168.

of these axes—according to the size of the plant—are slowly rotated, the wetted portions of the discs (about $\frac{1}{3}$ of the diameter) extracting moisture and heat from the air passing over them. This apparatus appears more complicated than the surface coolers just described, since it requires an agitator for the brine in the trough, besides the pulleys for driving the discs; it has, however, yielded very good results in practice.[1]

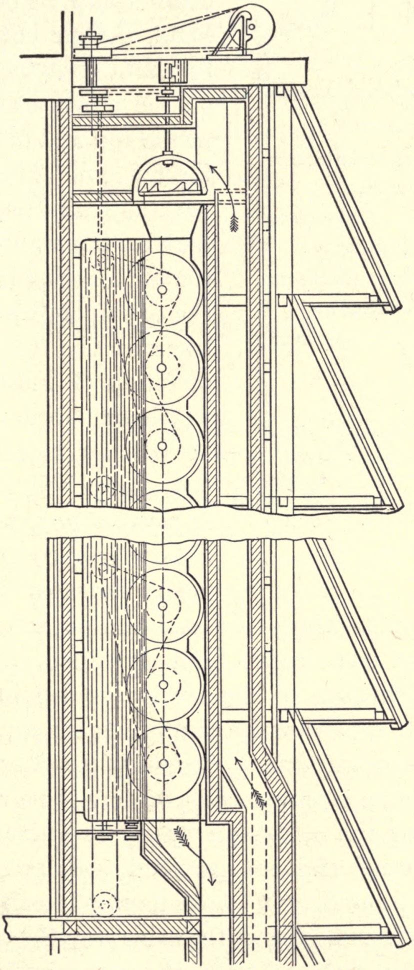

FIG. 169.

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[1] The work required by these air-coolers, when they are on a large scale,

This does not hold to the same extent for the *spray-coolers*, of which a diagrammatic sketch is given in Fig. 170. These have, however, the advantage that they admit of the ready production of a counter-current between, on the one hand, the air entering at the bottom and escaping at the top and, on the other, the brine flowing in at the top and falling in cascades to the bottom. The action of this apparatus really depends to some extent on the winding path taken by the air bringing it into contact with the dry under surface of the brine-basins. The frequent change in the direction of motion of the air, by which a uniform drying and cooling of the air is attained, introduces (especially when several sets of apparatus are employed) considerable resistance to the motion and hence increases the work to be expended in ventilating; the undesirable action of this on the cooling effect has already been explained. The change in the direction of motion of the air is avoided in the more recent forms of this apparatus by compelling the air to pass through a series of *cascades arranged one behind the other*, as shown in Fig. 171. The brine often carried away by the air can be prevented from entering the cold-room by plates which are placed in the pressure-channel and are provided with grooves for returning the brine deposited. The work required in all these apparatuses for continually pumping up the brine scarcely needs consideration in comparison with the work of ventilation; it must, however, be noted that such arrangements assume the existence of a

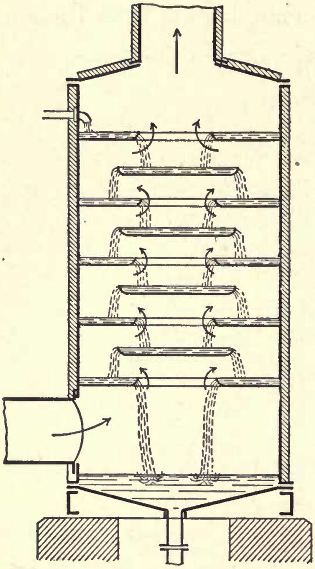

FIG. 170.

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has been found to be roughly 0·1 horse-power for a circulation of 1000 cubic yards of air per hour and a fall of temperature from 39° to 21° F.

brine-cooler (evaporator), through which the brine must first of all pass.

35. **The Cold Required for Air-cooling** depends partly on the objects in the cold-room, these having first to be cooled to the temperature of the room and then superficially dried. It is further influenced by the heat radiated and conducted into the room from outside, as this has to be prevented as far as possible from reaching the goods by the cooling of the air. A third source of heat is the high-temperature air coming in from the exterior; this is occasioned by the changing of the air from time to time necessary, by leaky walls, windows, or doors, and especially by the unavoidable opening of the doors, which are, however, often negligently left open. This evil may be obviated, on the one hand, by strict regulations,[1] and, on the other, by arranging the doors to open into an intermediate room, the doors of which should never be open at the same time as those of the cold-room. Leakage must be avoided by carefully stopping all cracks at the doors and windows (the latter must never be opened and should be made with at least double panes); the doors should not shut directly on to the door-posts, but on to pieces of felt nailed on to these. A fourth quantity of heat is introduced by the lighting of the room and by men staying in it.

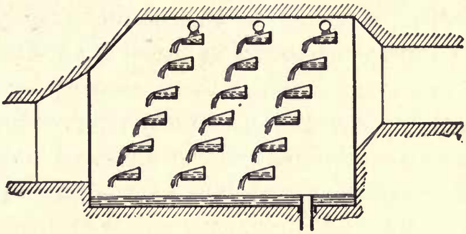

FIG. 171.

(*a*) The *heat to be withdrawn from the materials* placed in the cold-room is given by their weight, their mean specific heat, and the difference between their original temperature and that of the room. With meat and other material containing a large proportion of water, the specific heat varies from 0·7 to 0·8, i.e., it is very nearly that of water; the vigorous evaporation[1] which immediately begins at the surface of such bodies contributes very appreciably to a rapid fall of temperature, but is not to be taken into account in calculating the cooling necessary, since the vapour formed has to be condensed again in the cold-air plant. On the other hand, the complete cooling of the goods is spread over a longer period (according to *Zschocke*,[2] the cooling of a quarter of beef from 86° to 37° F. requires, roughly, 40 hours in a room the temperature of which varies from 43 to 37° F.), so that only a fractional part has to be allowed for per hour. If the goods are only brought in or removed once a day, approximately constant conditions may be reckoned on if the cooling of each fresh charge is distributed over 24 hours. With cold-stores connected, for example, with public slaughter-houses, this charge will amount approximately to one-half or two-thirds of the total contents, the capacity of the room being so calculated that, with a height of about 4 yards, 200–250 lbs. of meat occupy 1 square yard of floor surface.[3] If, for example, 50,000 lbs. of

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[1] Zeitschrift für Kälte-Industrie, 1896, page 4.

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[1] This phenomenon is the cause of the diminution in weight of meat so much feared by butchers. The manager of the slaughter-houses in Cologne, Mr. *Goltz*, found that after eight days, at 39° F.,

¼ an ox	weighing	156·5	lbs.	had	lost	7·5	lbs.
½ a pig	"	98	"	"	"	3·5	"
1 calf	"	84	"	"	"	7·5	"
1 sheep	"	78	"	"	"	3·5	"

while with warm dry outside air the losses were relatively somewhat greater. Dr. *Grassmann* found that, in the cold-store of the Royal Store Office at Thorn, the following percentage losses of weight occurred when the air temperature was from 23° to 26° F.:

	After 4 Months.	After 9 Months.
Beef	8·8	17·8
Pork	7·4	12·8
Mutton	11·5	23·4

For details of this matter, see Dr. O. *Schwarz*, "Bau, Einrichtung und Betrieb von öffentlichen Schlachthöfen" (Berlin, 1894), page 87.

[2] Deutsche Tierärztl. Wochenschr. IV, No. 29; see also Zeitschrift für Kälte-Industrie, 1897, page 17.

[3] Too great a height always causes waste of the cooling and makes it impossible to maintain a uniform temperature.

meat at 82° F. are introduced daily into a cold-store kept at 39° F., about 70,000 B.T.U. (50,000 multiplied by 0·8, the specific heat, and by (82−39) and divided by 24) must be withdrawn from the meat per hour in order to bring it down to the temperature of the store in 24 hours.

In the brewery, on the other hand, the temperature of the wort is brought down by the refrigerator to nearly that of the tun-room (39−46° or, on the average, 42·5° F.), so that very little heat has to be abstracted from it, while after it is introduced into the lager cellar it must be cooled to about 34° F.

(*b*) The *radiated and conducted heat* which enters the room depends, firstly, on the difference of temperature between the inner and outer air, and secondly, on the nature of the walls.[1] Of the greatest importance is the perfect dryness of the walls and floor, since any penetration of surface water or wetting through of the walls would cause an immediate loss of enormous stores of cold, besides vitiation (mustiness) of the inside air. If, therefore, the cold-room lies wholly or even partially in the region of the surface water, both the floor and the walls must be protected outside by a perfectly water-tight coating of *asphalt pitch.* Care must then be taken that the smell of asphalt has completely disappeared before goods are placed in the room, as otherwise they may be damaged thereby.[2] Further, it is advisable to lay the flooring material (clinker plates or, better, asphalt or concrete) on an *insulating layer* of *coke* or *wood ashes* 1¼−1½ feet thick, which, in its turn, should rest on a layer of *concrete* from 4 to 8 inches in thickness. The whole floor is to be so arranged that gutters with outlets are laid throughout at accessible places (the gangways).

The *walls* are fitted with a single or double insulating layer, 5 to 5½ inches in thickness, and are often made inside of porous or hollow bricks. The insulating space is never left empty, because, if this were done, circulation of the air would take place and loss of heat be favoured. As packing material, *ashes, cork chippings, kieselguhr, powdered chalk, leaf charcoal* (somewhat expensive), *loose pumice,* and *peat-dust* have given good results; less satisfactory is *sawdust,* which should always be mixed with kieselguhr or peat-dust.

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[1] See *Fischer*, "Studien über die industrielle Verwertung von Kälte, Civilingenieur, XXXVIII, Part 4.

[2] On this account, *Hess*, the manager of the slaughter-house at Straubing, advises the avoidance of asphalt and all tarry substances (Bayer. Industrie- und Gewerbebl., 1899, page 199).

So-called *slag-wool* should be avoided on account of the hydrogen sulphide formed by the partial decomposition it always undergoes.

The *inner coating of the walls* is of great importance, less, indeed, for its insulating effect than for its influence on the purity of the air and, consequently, on the preservation of the goods. According to investigations by Dr. *Popp,*[1] cement plastering is quite unsuitable, as it forms a good nutritive medium for micro-organisms; it is better to use a smooth, bright, waterproof coating (e.g., porcelain-enamel paint), which must be applied only after the walls are perfectly dry.

Where the *roof* is arched between carriers, it is sometimes insulated in a similar manner to the walls, i.e., by double arches, but for the sake of cheapness, a single arch of cement is generally used, covered with a layer of ashes or peat-dust 1–1½ feet thick.

Windows and fanlights,[2] which should always be double, ought never to be placed on the sunny sides (south and west); the same holds also for the *entrances.* The latter must be tightly closed by double-walled doors containing insulating material. Self-closing doors are to be highly recommended.

Special care is required in the insulation of *cold-stores on board ship,* especially when they are so large (in meat transports) as to reach the outer walls of the ship. Since these walls are constructed of iron- or steel-plate and are surrounded outside by rapidly moving water, the conditions are such as to favour the passage of heat to the inside. In order to limit this as much as possible, various means are employed; the wall of the ship is covered first with doubly varnished wood, then follows an insulating air-layer (often filled with sawdust or felt), beyond which are several layers of paper. Figures 172 and 173 show

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[1] Zeitschrift für Kälte-Industrie, 1897, page 4.

[2] According to investigations made by *Zsigmondy* (Wiedemann's Annalen der Physik und Chemie, New Series, vol. 49, pages 531 *et seq.*), green glass containing about 2 per cent of ferrous oxide admits of the passage of a considerable quantity of light, but stops nearly all the heat.

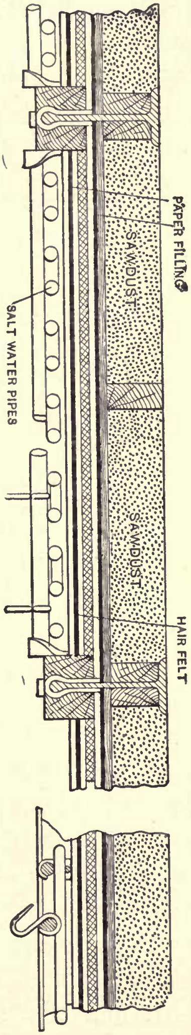

FIG. 172.

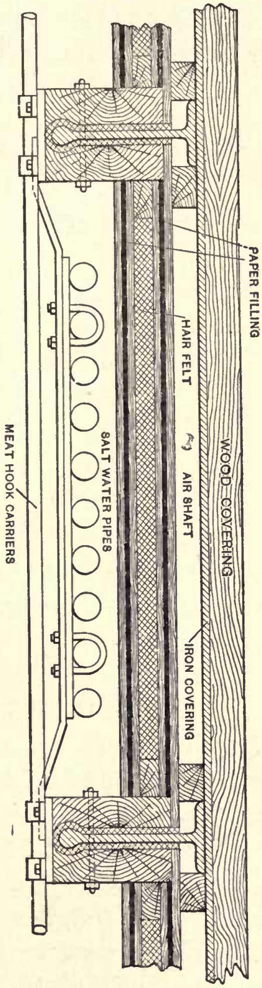

FIG. 173.

the method of insulating the cold-stores of the Cunard liners, "Campania" and "Lucania," in which the cooling is effected by means of brine-pipes.

The heat conductivity of the various insulating materials may be measured experimentally; the following table gives the number of B.T.U. transmitted per hour through 1 square foot of a layer of the material 1 foot thick.

TABLE XXI.

Material.	Heat Conductivity.	Weight per Cub. Ft. (Lbs.).
Wood ashes	0·040	
Pumice	0·044	23
Cork chippings	0·054	19
Powdered chalk	0·060	100
Slag-wool	0·068	
Leaf-wood charcoal	0·079	10–12
Kieselguhr (loose)	0·091	16–22
Cork	0·094	30–38
Powdered coke	0·107	22–34
Plaster	0·269	
Brickwork	0·470	92–106
Stonework	0·874	150–153

With the help of this table the *amounts of heat transmitted* per hour per square foot, with a temperature-difference on the two sides of 1° F., are found to be as follows:

(*a*) For a half-brick arch with a covering layer 1½ feet thick................0·04 to 0·06 B.T. U.

(*b*) For a wall three bricks thick with a double insulating layer 4 ins. thick. .0·08 to 0·12 B.T.U.

(*c*) For a stone-slab floor on a 1½-foot layer of coke ashes...............0·10 to 0·14 B.T.U.

For *double wooden doors* containing an insulating layer the corresponding amount of heat transmitted is 0·3–0·4 B.T.U., for *single windows* 1 B.T.U., and for *double windows* 0·5–0·6 B.T.U. In making a general estimate it is sufficient to take the value 0·16–0·2 B.T.U. per square foot of internal wall surface (including ceiling and floor).

Suppose, for instance, a cold-store has an inside length of

100 feet, a breadth of 50 feet, and a height of 11½ feet, that its walls, roof, and flooring are constructed as described above, and that it is lighted by side windows having a total area of 150 sq. ft. and has a double door 80 sq. ft. in area. Then if the outside temperature is 75° F., the temperature of the floor 54°, and that of the room itself 39°, the transmission of heat per hour will be:

For 5000 sq. ft. of flooring,	5000×15×0·14	=10,500	B.T.U.
“ 5000 “ “ “ roofing,	5000×36×0·6	=10,800	“
“ 3300 “ “ “ wall,	3300×36×0·12	=14,200	“
“ 150 “ “ “ window,	150×36×0·6	= 3,240	“
“ 80 “ “ “ door,	80×36×0·4	= 1,150	“
Total..............................		39,890	B.T.U.

In order to allow for faulty insulation and for unexpected rise in the temperature of the outside air, this amount should be increased by about 50 per cent, the value thus obtained being hence 60,000 B.T.U.

(*c*) *The cooling loss owing to renewal of the air* (ventilation) depends, like the heat penetrating from outside, on the difference between the internal and external temperatures, and, further, on the frequency with which the air is changed and on the moisture-content of the inside and outside air. In rooms containing food-stuffs which readily decay, such as meat, vegetables, and fish, the air should be completely renewed from four to six times per day; such renewal is, however, often very largely or entirely omitted on account of the attendant cooling losses. The consequence of this is that there is an unbearable smell in the store which not only renders it difficult to remain there but very soon becomes imparted to the stored goods and more or less lowers their value. When artificial circulation of the air is made use of, the removal of any quantity of air from the cold-room presents no difficulty, because a slight excess pressure can be maintained therein by means of the ventilators of the cold-air apparatus. The introduction of fresh air then necessitates the use of a special ventilator, but it becomes possible to introduce this air at the most suitable place, i.e., directly into the cold-air apparatus, while the waste air is removed by the suction-pipe just in front of the apparatus. The unsystematic arrangement of air-holes or flues in the cold-store is always to be deprecated, as it is then scarcely possible to control the renewal of the air, while the influence of the varying direction of the wind becomes more marked.

In order to maintain as far as possible constant conditions in the cold-store the renewal of the air should, of course, be continuous, but this is not prevented by regulating with traps or slides.

For calculating the cooling loss incurred by renewing the air, we shall go back to the above example, where the cold-room is taken as having a content of $100 \times 50 \times 11\frac{1}{2} = 57{,}500$ cub. ft., which would be sufficient for 50 tons of meat. The air escaping at about 38° and having a relative moisture-content of say 70 per cent, contains $0 \cdot 379 \times 0 \cdot 7 = 0 \cdot 265$ lb. of water-vapour per 1000 cub. ft., while the fresh air entering with a temperature of 75° and a moisture-content of 80 per cent carries with it $0 \cdot 8 \times 1 \cdot 339 = 1 \cdot 071$ lb. of water per 1000 cub. ft. The difference between these two amounts, namely, $1 \cdot 071 - 0 \cdot 265 = 0 \cdot 806$ lb., is the amount of water to be extracted from every 1000 cubic feet of the air, and, with an average heat of evaporation of 1100 B.T.U. per lb., this corresponds with a heat evolution of $0 \cdot 806 \times 1100 = 886 \cdot 6$ B.T.U. When the air is changed six times daily, i.e., 14,375 cub. ft. are introduced per hour, the cooling effect neutralised in this way is $14 \cdot 375 \times 886 \cdot 6 = 12{,}745$ B.T.U. per hour. Moreover, the air itself is to be cooled from 75° to 38°, and, taking the specific heat to be, roughly, 21 B.T.U. per 1000 cub. ft., this requires a further extraction of $14 \cdot 375 \times 37 \times 21 = 11{,}170$ B.T.U. per hour. Altogether, then, the renewal of the air requires the absorption of 23,915 B.T.U. per hour, which, in the case under consideration, is about half as great as the amount penetrating the walls from outside.

This loss has been considerably diminished of late years by the use of an *apparatus in which the waste air gives up the greater part of its store of cold*—which was formerly lost—to the fresh air. Assuming that this apparatus has an efficiency of about 0·5, approximately 12,000 B.T.U. are recovered in this way; the calculated loss in cooling effect must therefore be diminished by this amount. With efficient ventilation, however, this loss is quite great enough to make it worth while to utilise the smallness of the moisture-content of the escaping air in this apparatus, which consists simply of two channels separated by thin sheet metal.[1]

(*d*) The *loss incurred by opening the doors*, which is considerably diminished by making use of an intermediate chamber to prevent direct communication between the inner and outer air, scarcely admits of calculation and must be allowed for by increasing the total cooling effect by from 5 to 8 per cent.

(*e*) Finally, if the cold-room—for example, the fermenting- or lager-cellar in breweries—is to be artificially lighted and to have men working in it for any length of time, suitable allowances must be made for these. The following quantities of heat are developed per hour by:

1 man at work	about	520	B.T.U.
1 Argand burner consuming about 4 cub. ft. of gas	"	3600	"
1 16-candle-power incandescent electric lamp	"	120–160	"
1 wax candle	"	440	"

On this account, and also owing to the convenience of turning it on and off from outside, the electric lamp is to be preferred to all other methods of illumination.

For the example we have taken, the separate cooling effects are hence as follows:

(*a*) Lowering the temperature of the meat	70,000	B.T.U.
(*b*) Destroying the heat penetrating the walls	60,000	"
(*c*) Renewal of the air	12,000	"
(*d*) Losses by opening doors, etc.	12,000	"
Total cooling effect per hour	154,000	B.T.U.

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[1] Such an apparatus is described by R. *Stetefeld* in the Zeitschrift für Kälte-Industrie, 1900, page 68.

Let us suppose that the air is taken from the room at 38° and with a relative moisture-content of 70 per cent, so that 1000 cub. ft. contain 0·265 lb. of water-vapour, and led back at 26° and 70 per cent humidity, so that it contains $0 \cdot 242 \times 0 \cdot 7 = 0 \cdot 169$ lb. of moisture per 1000 cub. ft.; the lowering of the temperature by 12° will require $12 \times 21 = 252$ B.T.U. per 1000 cub. ft., while the drying uses up a further quantity of $(0 \cdot 265 - 0 \cdot 169)1100 = 106$ B.T.U., making a total of 358 B.T.U. Consequently the cold-air apparatus must supply 154,000 : 358 =, roughly, 430,000 cub. ft. of air per hour, which is about eight times the content of the room.

We are now in a position to determine the dimensions of the air-passages and the work required by ventilation. For a velocity of 16 feet per second, the cross-section of the main channel will have to be $\frac{430{,}000}{16 \times 3600} =$ about 7·5 sq. ft., and the ventilator work about 0·007 horse-power per 1000 cub. ft. of air, i.e., about 3 horse-power. Since the destruction of 1 horse-power liberates about 2540 B.T.U., which is transferred directly to the air, the cooling effect to be provided must be increased by $3 \times 2540 = 7620$ B.T.U. The total cooling effect for the example taken is hence, roughly, 162,000 B.T.U.

CHAPTER VIII.

MANUFACTURE OF ICE.

36. Laws of Ice-production.—The manufacture of artificial ice, which was used in place of natural ice for cooling purposes and is still employed in the household, formed the starting-point of the whole of the artificial-cooling industry. After it had been found that cooling could be carried out, not only in a more rational but also in a more effective way, without the use of ice, the production of the latter fell off to some extent, but it has lately been revived owing to the increasing demand for germ-free ice for the preparation of luxuries and to its use for certain industrial and sporting purposes (freezing process for boring shafts and for artificial skating-rinks). If it is a question of preparing *blocks* or *plates of ice* to be used elsewhere, the main condition is that the ice shall be readily removable from the so-called *generator* in which it is made; in other cases, where the ice has to work continuously on the spot, this condition no longer exists and, after solidification, the process is limited to replacing the ice used up. In the latter case, also, the purity of the product is of no account, while in the former, and especially when the ice is used for food, special arrangements are required.

The economical side of the question depends on the temperature of the water to be converted into ice, and also on the temperature to which the ice itself has to be cooled after being formed and on certain losses. As the formation of ice takes place practically always under atmospheric pressure, and therefore at 32° F.,[1] for 1 lb. always the same quantity of heat, namely,

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[1] The only exception is met with in the production of ice by the water-vapour cooling-machine mentioned at the beginning af Chapter II; here the

144 B.T.U., has to be abstracted. If, therefore, the water has an initial temperature of 50° F., and the ice, which has the specific heat 0·5, has to be cooled to 23°, the cooling required is theoretically $18+144+0{\cdot}5\times9=166{\cdot}5$ B.T.U. With water at 59° and an ice temperature of 14°, this amount rises to, roughly, 175 B.T.U.

As is very evident, in the making of *block and plate ice* the losses mentioned above became increased both with those parts of the surface of the water and subsequently formed ice which are subjected to radiation and conduction from outside and also with the time taken by the cooling and freezing processes. This time will, however, be *shortened* by *increasing the ratio which the surface of the water exposed to the cooling action bears to its volume.* This ratio should, therefore, be chosen as large as possible, i.e., the *ice blocks* should be made quite thin and flat, so that the freezing may be rapidly effected by the external cooling action. But if this is carried too far, the unavoidable melting losses in extraction (which takes place by thawing, as we shall see later) and during transport—which with blocks of 56 lbs. amount to 6–8 per cent and with those of 28 lbs. to 9–12 per cent—become very considerable, so that it is advisable to keep within moderate limits. In practice it has been found that if the volume of the block is given in cubic yards and the surface exposed to the cooling action in square yards, the ratio of the latter to the former is suitably taken as between 29 and 35,[1] when the period of

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finely divided cold, fresh water is introduced in such quantity that a part of it (about one-eighth) evaporates in the vacuum, while the rest (seven-eighths) forms a thin layer on the walls of the vessel (ice-cell), where it freezes. Since this process is carried out without the help of any intermediate substance, it should theoretically give the greatest yield; as is shown by Table IV, for each indicated horse-power of the water-vapour cooling-machine, roughly 20,000 B.T.U. are removed, corresponding with the production of about 112 lbs. of ice at 14°. Unfortunately this has to be increased by the work of the air-pump and certain accessory apparatus, which have an unfavourable influence on the yield. But as the ice obtained in this way is very dense and clear, the method might very well be further improved.

[1] This number, which is of great importance in the designing of ice-generators, was calculated by Professor *Fischer* (Dresden) from detailed experiments.

freezing of a block under constant conditions of temperature increases approximately as its volume. For instance, blocks of 56 lbs. weight and with a surface area of 0·096 sq. yd. require about 30 hours, and those of 28 lbs. weight and surface area 0·048 sq. yd., only 12–15 hours for their formation; taking 180 B.T.U. as used per lb., this corresponds with a mean hourly transmission of, roughly, 350 B.T.U. through each square yard of surface. It is here assumed that the thin-walled metal freezing-vessel (*ice-cell*) is cooled outside by cold brine.

The whole procedure is such that in a short time after the starting of a vigorous natural motion in the vessel the water first cools to 32° F. and then freezes in layers from the outside, at first quickly and afterwards more slowly, the temperature of the outer frozen layers gradually falling more and more, while the inner liquid portion remains at 32° until it solidifies. This explains the relatively long time taken in freezing thick blocks.

This value is still more unfavourable in the preparation of *plates of ice*, which are generally formed in a water-bath by freezing at two side cooling surfaces of very large dimensions, the thickness being about 8 or 12 inches. When the plates have a parallelepiped form, the ratio of the surface exposed to the cooling action to the total volume may, therefore, be not greater than 2·9–3·3, because only one side-face is to be regarded as active. Comparing this with the corresponding ratio (about ten times as great) for blocks of the same thickness (of about 56 lbs. weight), it will be seen that about ten times as long must be allowed for the freezing of the plates, which often extends over 10 or 12 days. The consequence is that ice-generators for producing plates have enormous dimensions compared with those for blocks and require correspondingly more expensive installations.

The production of ice is usually effected, as already remarked, by means of a salt solution, which extracts heat from the water and gives it up to the evaporator coils of a cooling-machine. The *work used up* by the latter is the greater the lower the temperature to which the evaporator has to fall in order to cool down the ice formed. If, for example, ice at 23° F. has to be prepared, the temperature in the evaporator coils must be about 14° F., whilst the previous cooling of the water itself from about 50° to 32° would only require an evaporator temperature of 29° or 27°. On this account it seems convenient, at any rate for ice manufacture, *to separate the cooling of the water completely from the production of ice,* i.e., to carry it out in a special, smaller machine before filling the freezing-vessels. This may be done especially simply by taking the water from the fresh-water cooler, provided that this is served by a separate cooling-machine. It would, however, be impracticable to cool the ice at a place other than that at which it is formed, because any possible small saving in work would be far more than compensated by the losses occurring during the transport.

The production of block ice under these conditions (i.e., using brine as a cooling medium) requires an evaporator temperature of from 14° to 5°; in making plate ice, the freezing is sometimes accelerated by working at $-4°$, but this naturally increases the work expended. To completely determine the latter for a given yield of ice it is necessary to take into account a further loss, which comes into play only in the production of blocks, namely, the effect of the cooling of the cells and their carriers on the temperature of the ice. These cells always consist of rectangular vessels of fluted galvanised iron open at the top; they vary from 2 to 3½ feet in length, contain 28 lbs. or 56 lbs. of ice, and weigh about 22 lbs. or 37 lbs. respectively, while the carrier weighs about 3 lbs. per cell in the first case and about 2 lbs. in the second. The specific heat of the material of the cells and carriers has an average value of 0·12. With plate ice, which is mostly sawn from the cooling surface and taken straight from the generator, such losses do not occur. Hence, for the production of 1 ton of ice at 21° F. per hour from water at 59° F., we have the following data (Table XXII).

The cold used up hence varies only inconsiderably with the shape of the ice produced, so that for large installations we may take it as, roughly, 200 B.T.U. per pound of ice, and for smaller

TABLE XXII.

Shape of the ice	Block Ice.		Plate Ice.
Weight of each mass of ice (lbs.)	28	56	4480–5600
“ “ cells and carriers per ton of ice (lbs.)	2,000	1,560	
Cold required for cooling and freezing, B.T.U.	395,000	395,000	395,000
“ “ “ replacing losses due to melting, B.T.U.	47,400	31,600	8,000
“ “ “ cooling the cells and carriers, B.T.U.	9,120	7,110	
“ “ “ annulling heat penetrating from outside,[1] 5, 7·5, and 10 per cent, B.T.U.	19,750	29,625	39,500
Total cold required for 1 ton of ice net, B.T.U	471,270	463,335	442,500

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[1] Concerning this quantity there are very few experimental results, besides which it varies considerably in one and the same installation; see *Guthermuth* and *Salomon*, "Versuche an einer Pictetschen Eismaschinenanlage," Zeitschrift des Vereins der Ingenieure, 1889.

The above value, 10 per cent, for plate ice must be regarded only as an estimate and must, in view of the large dimensions of the generators, be often exceeded.

ones, owing to the relatively greater losses, as 220 B.T.U. per pound. Taking on an average 16,000 B.T.U. per indicated horse-power of the compressor, or, including the work of agitating and pumping, 10,000 B.T.U. per effective horse-power, we see that for the former an hourly ice production of 70–80 lbs. is possible, and per effective horse-power 46–50 lbs., under normal working conditions.

The *work required for extracting the ice* may be arrived at from the fact that the cells together with their frames have to be lifted on an average 5 feet, once from the generator and once from the thawing-off tank. Taking the weight of the cells and frame for 1 ton of ice as 2000 lbs., this requires $2\times5\times4240=$ 42,400 ft.-lbs., which, when a block and tackle are employed, is increased to at least double, i.e., to 84,800 ft.-lbs. To this must be added the work required for moving the cell-frames backwards and forwards, for placing the plank cover on the generator, for putting into action the filling and tipping appliances; this amounts to about 7500 ft.-lbs., so that altogether 92,300 ft.-lbs. have to be supplied as hand-work. This represents approximately the average amount of work done by a strong workman, who would therefore be kept completely occupied with the work of a 1-ton generator. Hence, where the engineer of the cooling-machine has also to attend to the ice-generator, it is necessary to make use of cranes, etc. (see below), worked by power.

37. Ice-production Plants are, almost without exception, connected with the evaporator of the cooling-machine, and this is also mostly the case when the ice-making uses up only a part of the total cooling effect produced. The so-called *ice-generator* serves in this case as a brine-cooler as well, the lowest temperature of the brine being made use of immediately for the production of ice. Where large quantities of ice are to be made this combination is not very economical, because the whole of the brine, which otherwise would have to be cooled to 22–23°, must be brought to about 18°, the work being thus considerably increased.

Figs. 174 and 175 represent, in its simplest form, a plant for the *production of block ice* by Linde's method. *A* is the compressor, *B* the driving-engine, *DD* the main leading to the condenser *K*, *R* the regulating-valve in the intermediate main *CC*, and *G* the ice-generator which contains the evaporation tubes and consists of a well-insulated box covered at the top with stiff planks.

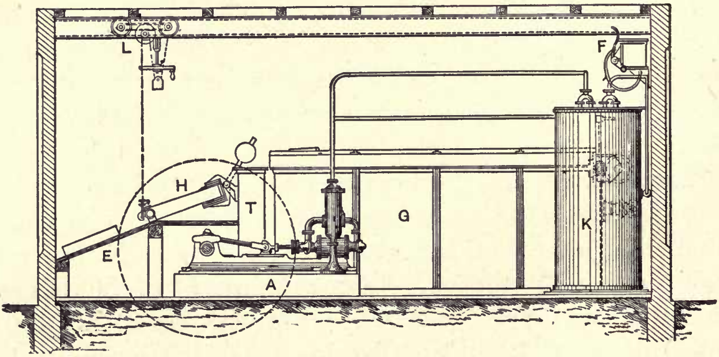

Fig. 174.

In Fig. 175 the plank cover is removed at the front and back ends of the generator, so as to render visible the ice-cells *z* hanging in their carriers. Hooks are fitted at the ends of the carriers to allow the latter to be raised, moved sideways, and lowered by a travelling-crane. When a series of cells are frozen through, the planking above is lifted and the cells removed, brought over a vessel, *T*, at the end, filled with lukewarm water and immersed for a few minutes in it. In this way the blocks become loosened from the cell-walls, and, when the cells are brought into the position shown, by means of the tipping arrangement *H*, the ice slides on to the inclined plane *E*, from which it may be transferred either to the place where it is to be used or to the ice-store. The emptied cells are raised by the travelling-crane, carried to the other end of the generator and again filled with water from the vessel *T*, after which they are returned to their place in the generator, where the freezing

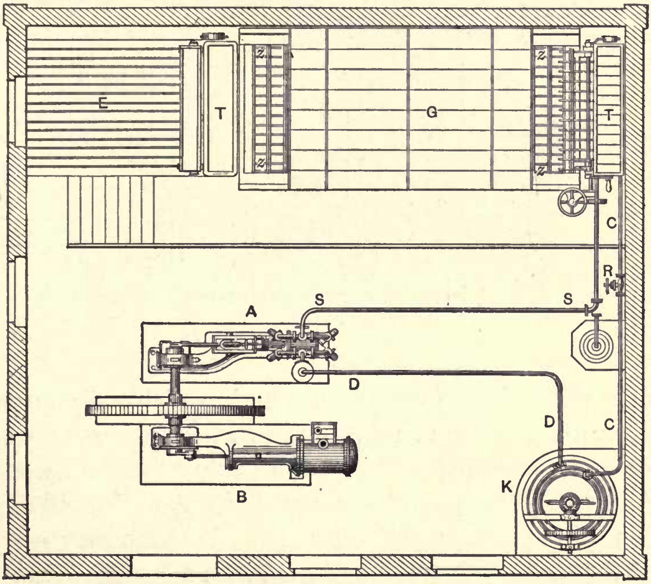

FIG. 175.

process begins anew. Every series of cells is treated in this way in a definite interval of time.

To every cell in the series there corresponds an exit in the tank *F*, this being mostly a bent tube which is fixed to a horizontal one and rotates with it and is connected with the filling vessel by means of a rubber tube. In Figs. 174 and 176 these tubes are shown in the upper position, in which their open ends are above the surface of the liquid in *F*, so that even without taps no water can flow out. This can only take place when the tubes are turned over and the streams of water then fall directly into the cells in front. In order that the tubes may not turn over of themselves, they are provided with a counterpoise. The cells are prevented from becoming too full—which might lead to the entrance of water into the generator—by means of an overflow leading back to the tank, which is divided by cross-walls into a number of equal compartments, so that the cells may be filled equally.

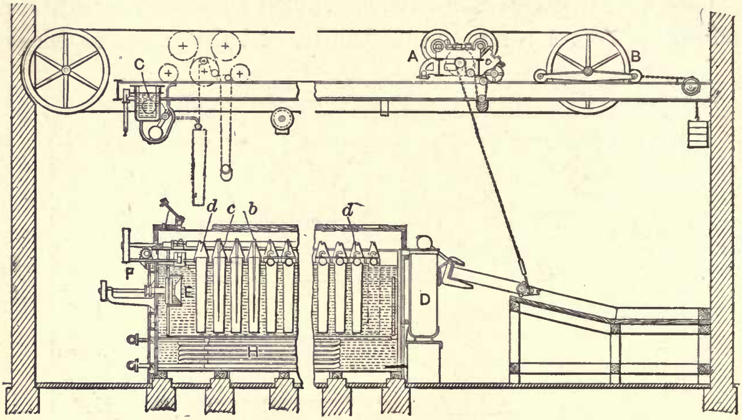

FIG. 176.

The generator is usually divided into two parts: in the lower space, covered at the top by sheet-iron, are placed (Figs. 176 and 178[1]) the evaporator coils surrounded by the brine, which is forced by a pump into the upper space containing the ice-cells, and, after circulating there and becoming warmed, falls at the other end into the lower space to be cooled again. The consequence of this arrangement is that the generator has to be made moderately tall, while there is a possibility of large losses of cold through the surfaces of the lower and coldest space. Further, the time required for the cells to freeze increases with the temperature of the brine, i.e., with the distance of the cells from the circulating-pump.

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[1] See *Diesel*, "Linde's Refrigerating Machines and Cooling Arrangements," Zeitschrift des Vereins der Ingenieure, 1893.

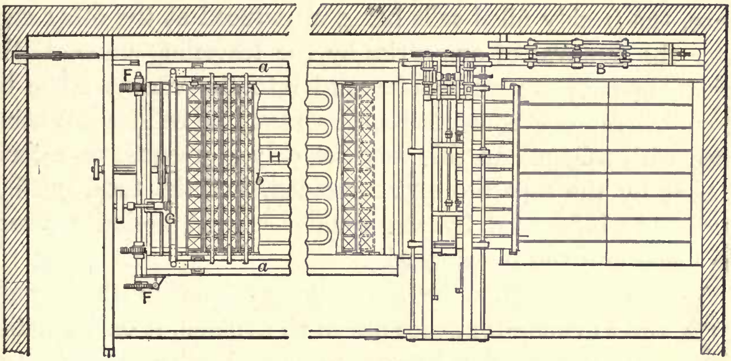

FIG. 177.

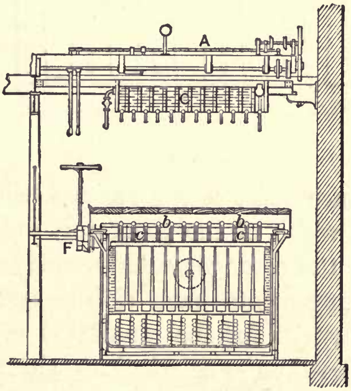

FIG. 178.

In Linde's construction, shown in Figs. 176–178, this defect is remedied by removing the series of frozen cells from the end of the generator where the cold solution enters from the lower space, the freshly filled cells being introduced at the other end. After the removal of one series by the travelling-crane *A*, the remaining ones are pushed forward into the vacant space by an endless screw *F* connected with a rack, the cell-frames being fitted with rollers. In this way not only is unequal extraction of heat by the separate series avoided, but the total yield is favourably influenced by the counter-current action between the brine and the freezing-cells.

If at the same time the losses taking place through the walls of the lower evaporator space are to be avoided, it only remains to place the evaporation tubes themselves between the separate cells, as is more generally done in America (see Fig. 179). In this case the distance apart of the various cells in a series, and therefore also the width of the generator, must be somewhat greater than with the arrangement described above.

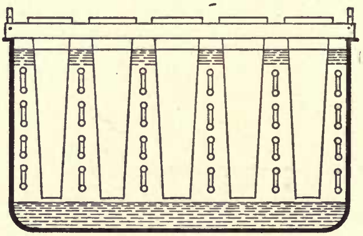

FIG. 179.

The above arrangements have also been lately introduced into chemical industries for the separation of crystalline substances from mother-liquors or solutions, since they have been found to be simpler and more economical than the older rotating apparatus,[1] mostly fashioned after the so-called hand ice-machines. In the brown-coal industry in Saxony, ordinary ice-generators with cells (of circular cross-section) have given good results in the separation of paraffin from the so-called lubricating oils, the brine temperature being about 14° F.; the solidified mass is several times submitted to pressure after removal from the cells.

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[1] *Scheithauer*, Die Fabrikation der Mineralöle, etc., Brunswick, 1895, pages 176 *et seq.*

Professor *Lunge* of Zürich has recently patented a very similar process for the separation of *monohydrated sulphuric acid crystals* [1] from concentrated sulphuric acid by cooling to about −4° F. in the generator cells. The complete separation of the crystals from the adhering mother-liquor is subsequently effected by centrifugation.

In this connection may be mentioned also *Walfard's disgorging process*, which was successfully introduced into the manufacture of effervescing wines by the Linde Ice-Machine Company and which allows of disgorging or removal of the deposit formed during fermentation in bottle, without much loss of the contents. After shaking, the bottles with the stoppers—on which the deposit has collected—downwards are immersed an inch or so deep in a cooling-bath (alcohol or glycerine) at about 4° F., so that the wine is frozen, forming a small ice-stopper containing the deposit. On removing the bottles and placing them upright, the cork, deposit, and ice are blown out together by the carbon dioxide in the bottle.

On *ships* the ice-generators must be so arranged that the movement of the ship causes neither spurting of the brine or water nor mixing of these two liquids. Linde's Ice-Machine Company, who have in the last few years successfully erected many cooling installations on board ship, have given to their generators the form represented in Figs. 180 and 181.[2] The cells are enclosed in a special space, but are placed not immediately into this but into pockets having as nearly as possible the same shape as the cells. Round these pockets circulates brine cooled in the lower space by means of ammonia and kept in motion by an agitator. Both these spaces are shut off tight from the outside. After introduction into the pockets, the cells are closed by loosely fitting covers, which are pressed tightly down by the outer cover.

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[1] The complete description of such an apparatus, with views, is given by the maker, W. *Kaufmann*, in the Zeitschrift für Kälteindustrie, 1901, page 2

[2] See *Habermann*, Die Kühling auf Schiffen, Zeitschrift für Kälte-Industrie, page 89.

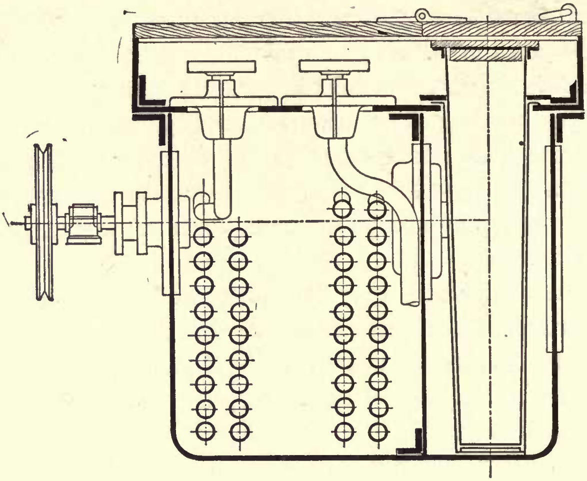

FIG. 180.

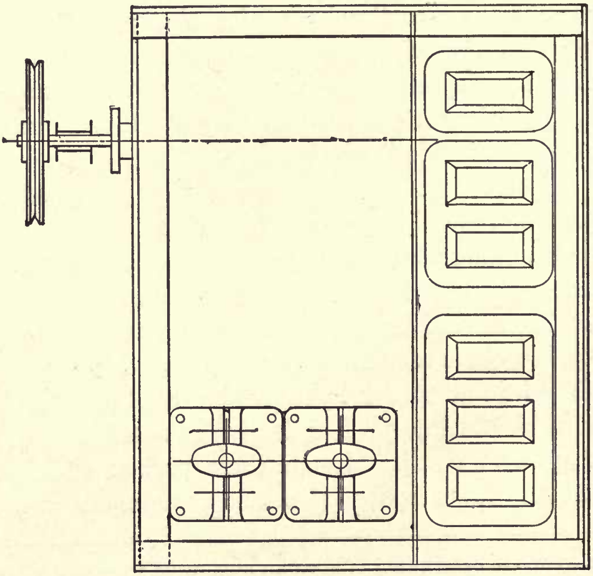

FIG. 181.

The *manufacture of plate ice*, which is seldom carried out in Europe, requires the construction of freezing elements of the same flat area as the desired plates (Fig. 182). The elements themselves are thin-walled, flat boxes which are surrounded by the evaporator-tube systems either by themselves (*Pusey* system) or immersed in brine (*Smith* system). In either case the plates are removed by a hand- or power-saw or else by means of thawing, the freezing element being placed for a while in connection with the condenser of the cooling-machine.[1]

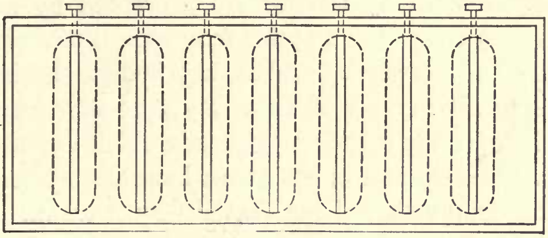

FIG. 182.

The ice prepared by the methods described above contains, of course, all the impurities of the water and, in particular, is rendered turbid by a large number of small air-bubbles which escape from the water on freezing; it is hence salled *opaque ice.*

Its specific gravity has the average value 0·85. The impurities of this ice, including numerous bacteria, which come from the air, render it unsuitable and even dangerous for use in foods. Further, the resistance it offers to atmospheric influences is but slight and its transportability hence limited. These properties are possessed in a much higher degree by *crystal ice* or *clear ice*, the use of which has greatly increased in recent years. Owing to the absence of air-bubbles this form of ice has a specific gravity much greater than that of opaque ice, namely, about 0·92.

The *manufacture of clear ice* varies according to whether only the air-bubbles and the consequent opacity are to be avoided or really *germ-free* ice has to be prepared. In the former case the cells are filled with well- or tap-water and 9 ozs. of alum added per 100 gallons of water, while the formation of air-bubbles and their retention by the ice are prevented by an agitator placed in each cell. The agitator may consist of a plate moved backwards and forwards, of hanging bodies or chains, etc.[1] Among the vast number of these, an especially simple form is that of Linde, consisting of vertical rods which dip into the cells (see Figs. 176–178) and in their turn hang from cross-pieces on an agitator frame moved horizontally by a crank. Since the two side carriers *a* of this frame rotate about vertical spindles placed at the other end of the generator, the amplitude of vibration of the cross-pieces with their rods becomes smaller as they approach the spindles. Where the cells move towards the corresponding end of the generator during the freezing process, the amount of movement of the stirring-rods diminishes with the quantity of water remaining in the middle of the cell. About 10 or 12 per cent of the water is left in the cell after the rod is removed, and this becomes opaque when frozen; but, if necessary, this water can be removed and replaced by distilled water.

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[1] Details on the manufacture of plate ice are given in the work of *Gutermuth*, "Amerikanische Eiswerke," Zeitschrift des Vereins der Ingenieure, 1894.

The rods must, of course, be taken out every time the cells are moved, i.e., at every fresh filling of the cells. The whole process thus becomes very involved.

For preparing germ-free ice, distilled water is exclusively used. In the ordinary case, where the cooling-machine is driven by steam, the *condensed steam*—which must be condensed in a surface condenser to avoid the absorption of air [2]—may be employed, the water being purified from oil and other impurities by *sand-* and *charcoal-filters*. This method can always be adopted when the manufacture of ice represents only a fraction of the total work of the installation, since, as we saw above, 1 indicated steam horse-power corresponds roughly with 45 lbs. of ice per hour, while the consumption of steam is about 22 lbs. for a good single-acting engine and only about 18 lbs. for a compound engine. Hence, when the work of the installation is used exclusively or mainly for ice-making, the remainder of the distilled water required must be provided for by other means. It would be very uneconomical if for this purpose water were simply evaporated in a steam-boiler and liquefied again in a condenser, since in this way only 6 or 8 lbs. of water would be obtained per lb. of coal.

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[1] It is here quite impossible to describe in detail the large number of such arrangements, which are often very unpractical.

[2] The production of crystal ice is carried out most simply by means of an absorption machine, from the heater of which a sufficient supply of condensed and uncontaminated water can be obtained (see Chapter V).

On the other hand, the steam thus formed can (see Fig. 183) be so condensed in a second boiler, K_2, that in the latter fresh water—previously warmed in an interchanger or economiser, A_2, by the heat of the condensed liquid—entering at C_2 will be evaporated; the steam obtained in this way can be similarly used in a third boiler, and so on, any quantity of distilled water being thus procurable, at all events theoretically, for a very small outlay of fuel.[1] It has been found in practice that, in order to overcome the resistance to the motion of the steam, a difference of pressure of from 7 to 14 lbs. per square inch, or, on an average, 10·5 lbs., must exist between two consecutive boilers, so

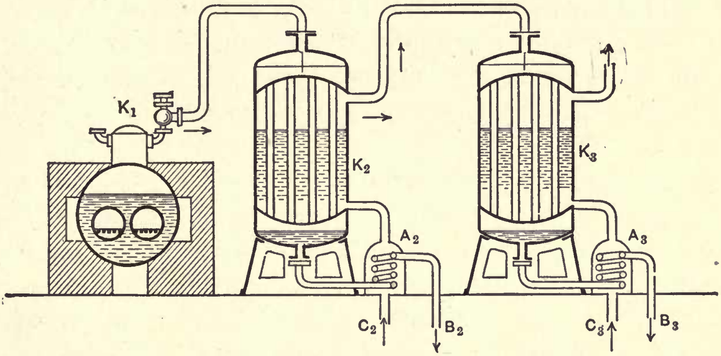

FIG. 183.

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[1] Such methods are largely employed in chemical industries (sugar and spirit manufacture).

that with an initial steam pressure of 70 lbs. per square inch, an efficiency of 80 per cent in each apparatus, and an evaporation in the primary boiler of 7 lbs. of water per pound of coal we may expect to obtain $7+0\cdot8\times7+0\cdot8^2\times7+0\cdot8^3\times7+0\cdot8^4\times7+0\cdot8^5\times7=25\cdot68$ lbs. of distilled water per lb. of coal where the evaporation is carried on in six steps. It must, however, not be overlooked that each of the boilers must have approximately the same heating surface as the primary boiler K_1 heated with coal, and that, further, the distillate obtained at B_2, B_3, etc., must first of all be submitted to the action of an air-pump. The plant will hence prove an expensive one, so that, as in the Linde Company's method,[1] it is well to use only one such secondary boiler, inserted either between the main boiler and the steam-engine or between the latter and the condenser; in this way, when the purified condensed steam is also used, the total distillate amounts to about 1·8 times the quantity of steam passing through the engine.

Since completely de-aërated water absorbs air with avidity, especially if strongly cooled, great care must be taken that all the mains leading from the distillation vessel to the filling apparatus are made perfectly tight. In order to avoid, as far as possible, absorption of air during the filling, the pipes *A* (see Fig. 184), which dip into the separate cells *Z* and are connected by india-rubber pipes with the filling vessel, must be long enough to reach nearly to the bottoms of the cells; this avoids almost entirely the formation of a free jet. Further, to prevent air from collecting in the pipes *A* and the distributing pipe *F*, the pipes *A* are furnished with foot-valves, *V*, which only open by being thrust against the bottom of the cell and close immediately the filling arrangement is raised.

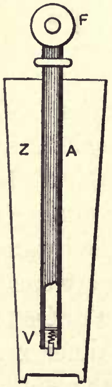

Fig. 184.

38. The Freezing Process as Applied to Shaft-boring and to the Preparation of Artificial Skating-rinks differs from the above-described manufacture of ice in that it requires the preparation of ice to be used where it is formed, so that it need not be melted away from the cold surfaces to which it is frozen fast. The working is hence much simpler than in other applications of artificial cold, but the arrangement of the plant, at all events in the first case, presents relatively great difficulties. The *freezing process*, first proposed by *Pötsch* in 1886 for sinking shafts in boggy rock and since then used with success in various localities, consists in forming—in watery soil or rock strata—solid ice to resist the pressure and prevent the infiltration of the external water, and in maintaining this until the dangerous layers are traversed and the shaft protected from water by the insertion of a braced iron pipe (so-called tubing), outside of which concrete and quick-setting cement are stamped in. The ice in question, which is either cylindrical or annular in form with a perpendicular axis (coinciding with that of the shaft), is gradually formed by the freezing of the water in the soil round a series of double tubes arranged in a circle having a diameter about 5 feet greater than the shaft at a distance apart of about 3 feet; in these pipes brine from the evaporator of the cooling-machine circulates.[1] These tubes can, of course,

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[1] Details are given in the paper by *Diesel*, referred to above.

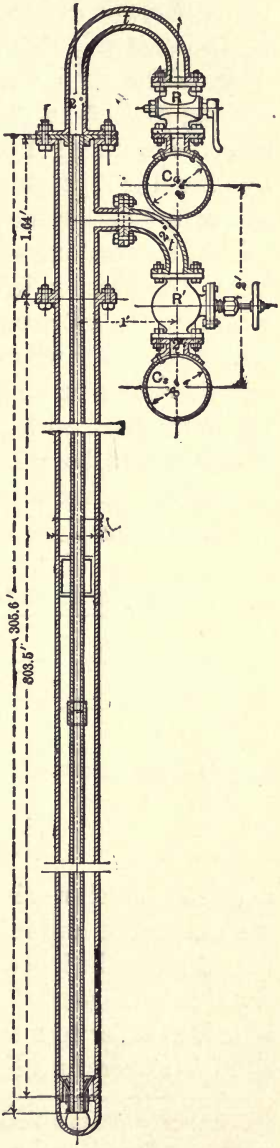

Fig. 185.

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[1] Proposals to allow the cooling-agent itself to circulate in the pipes and

only be introduced after the necessary holes have been bored to the required depth and are then connected. They always consist of the best steel; the outer ones mostly have a bore of from 4 to 5 inches and the inner ones of 1 inch, the thickness of the walls varying from $\frac{9}{32}$ to $\frac{5}{32}$ inch; the exterior cooling surface hence amounts to from 1·1 to 1·3 sq. ft. per foot run. As is seen from Fig. 185, the separate lengths of the outer tube are screwed together by means of internal union couplings and those of the inner tube by external union couplings; the joints are rendered tight by hemp string soaked in tar and red lead. The inner tube is kept central to the outer one by means of sleeves furnished with ribs arranged at intervals of about 30 yards, and at the end is fastened centrally in the hemispherical end-piece of the outer tube, which is also provided with ribs and into which the inner tube opens. Above ground, both the inner and outer tubes are joined, by way of suitable connections and cut-off arrangements, with annular collecting tubes C_a and C_s of about 8 inches bore (see also Figs. 186 and 187), which receive the brine through a main leading from the evaporator and return it through a second pipe after it has passed down the inner tube to the bottom and up the outer one. By this arrangement the *coldest brine acts at the lowest place,* where, in order to meet increase of hydraulic pressure and also rise of temperature as the shaft deepens, the ice-jacket must

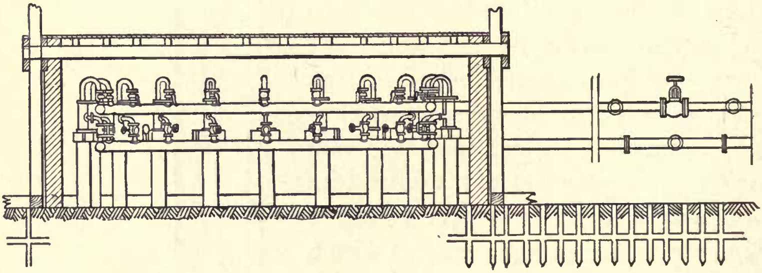

Fig. 186.

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thus not only obtain a lower temperature but also avoid any leakage of brine which would dissolve the ice formed, have not yet proceeded further than the experimental stage. Such procedure would also be hindered by the difficulty which the evaporation causes in maintaining the circulation.

have its greatest thickness. The latter naturally depends also on the nature and water-content of the rock, these being ascertained during the boring of the holes for the freezing-tubes. The velocity of the brine in the outer tubes is never allowed to exceed 5 inches per second in order not to increase uselessly the expenditure of energy by friction and also to admit of the cold being completely made use of; in the inner tube, however, the solution is carried as quickly as possible to the place where its action is to be exerted, its velocity here being 5 or 6 feet per second. When the mean temperature of the shaft is 50–54° F., the brine is cooled in the machine to 5–0° F.; it then leaves the freezing-pipes again at from 8° to 2° F.

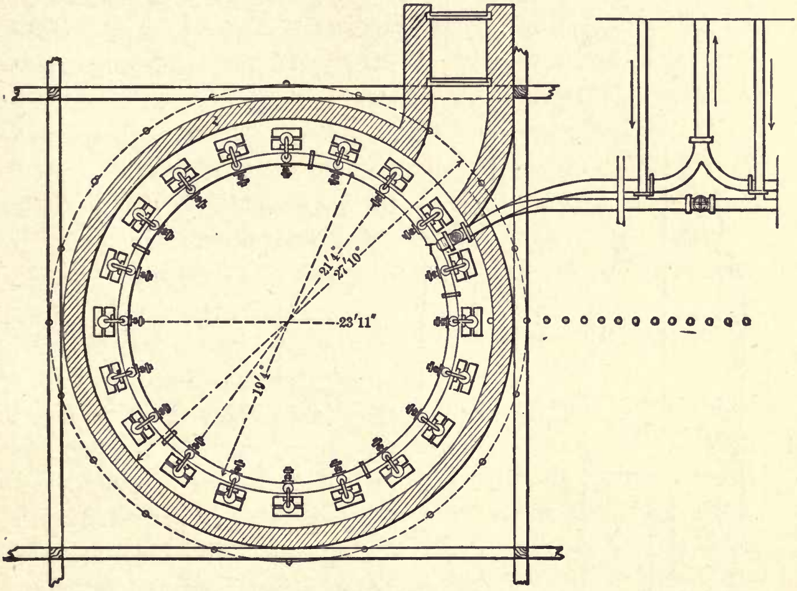

FIG. 187.

After the plant is fixed, the procedure is such that the material surrounding the pipes both on the inside and outside becomes rapidly cooled. As soon as that in immediate contact with the pipes reaches the freezing-point, ice begins to form and gradually spreads concentrically from the pipes, which absorb, on an average, 85–90 B.T.U. per square foot per hour. Under normal conditions it may be expected that the ice spreads for 18 inches outside the pipes and for a yard inside them, its temperature falling gradually from the pipes to the inner and outer edges of the ice, where it is 32° F. The earth is further somewhat cooled to a distance of about five feet, beyond which the temperature is normal. After these conditions have been reached, it is only necessary to make up the cooling losses during the boring, these amounting to from 18 to 20 B.T.U. per hour per square foot of the total inner and outer surface of the ice-jacket. Only after the boring and tubbing are complete is it safe to remove the brine from the cooling-pipes, to remove these from their holes and fill the latter with concrete. If the brine is not to be used again the simplest plan is to pump warm water into the pipes, which thus become rapidly thawed. When it is required for subsequent use, the brine is raised by means of cylindrical tubes which are provided with foot-valves (Fig. 188) and are about 5 feet long and have an external diameter (about $\frac{5}{8}$ inch) less than the internal diameter of the connecting sockets.[1]

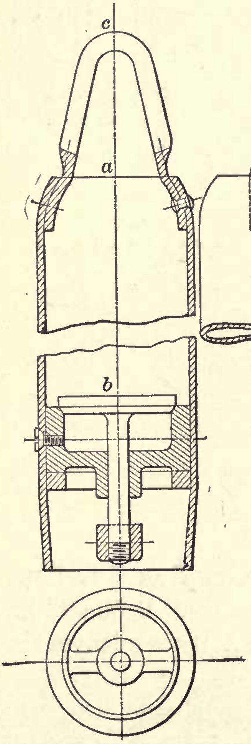

FIG. 188.

Suppose, for example, we have to sink a shaft 15 feet in diameter to a depth of 150 feet, the rock containing on an average 20 lbs. of water per cubic foot, while a layer of sandstone 30 ft. thick carries 40 lbs. of water per cubic foot; further, let us take the specific gravity of the anhydrous rock as 2·3 and that of the dry sand as 1·4, the specific heat of the two materials in the dry state 0·2 and the temperature of the ground as 50° F. First of all we arrange about 20 freezing-tubes in a circle of 20 feet diameter, around which there forms an annular block of ice having an inner diameter of about 13 feet and an outer one of 23 feet; the volume of this block for a depth of 150 feet is hence about 42,400 cub. ft. Under the conditions given above the weight of the water-content amounts to 1,017,600 lbs. and that of the rock and sand to 5,608,000 lbs. Both these masses are cooled to 32° F. before freezing and are afterwards brought to a mean temperature of $\frac{1}{2}(32+4)°=8°$, assuming that the temperature rises uniformly from the pipes to the edges of the mantle. The space within this ring, which is cooled to a mean temperature of $\frac{1}{2}(32+50)°=41°$, has a content of 19,900 cub. ft. and consists of 477,600 lbs. of water and 2,633,000 lbs. of rock; while outside the ice is a hollow cylinder which has to be cooled to this same temperature (41°) and which has an outer diameter of 37 feet, an inner one of 23 feet, and measures 98,960 cub. ft., comprising 2,375,000 lbs. of water and 13,090,000 lbs. of rock and sand. Finally, owing to the strongest cooling action taking place at the bottom, the lowest part of the inner space is cooled to 32° F. and entirely frozen for a depth of about 30 feet, by which 100,000 lbs. of water and 560,000 lbs. of rock and sand are affected.

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[1] Details of this process are given in the work of *Saclier and Waymel*, "Fonçage des puits de Vicq par le procédé Poetsch," Bulletin de la société

We have therefore to supply the following amounts of cold:

(*a*) For cooling the water and rock of the hollow cylinder

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minérale de St. Etienne, 1895; an extract is given in the Zeitschrift für Kälte-Industrie, 1896. This publication contains the results of very extended investigations, from which the experimental values given in the text are mainly taken.

Interesting information on this subject is also given in the comprehensive work of F. Schmidt (Paris), "Die Benutzung des Gefrierverfahrens zur Ausführung bergmännischer Arbeiten," Zeitschrift für Kälte-Industrie, 1898.

from 50° to 32°, $1{,}017{,}600 \times 18 + 5{,}608{,}000 \times 18 \times 0{\cdot}2 =$ 18,317,000 B.T.U.

(*b*) For freezing the water at 0°, $1{,}017{,}600 \times 144 =$ 146,530,000 B.T.U.

(*c*) For the further cooling of the ice and rock to 8°, $1{,}017{,}600 \times 24 \times 0{\cdot}5 + 5{,}608{,}000 \times 24 \times 0{\cdot}2 = 26{,}918{,}000$ B.T.U.

(*d*) For cooling the inner cylinder from 50° to 41°, $477{,}600 \times 9 + 2{,}633{,}000 \times 9 \times 0{\cdot}2 = 4{,}739{,}800$ B.T.U.

(*e*) For cooling the outer cylinder from 50° to 41°, $2{,}375{,}000 \times 9 + 13{,}090{,}000 \times 9 \times 0{\cdot}2 = 23{,}562{,}000$ B.T.U.

(*f*) For cooling the lowest part of the inner space from 41° to 32° and freezing the water it contains, $100{,}000 \times 9 + 560{,}000 \times 9 + 100{,}000 \times 144 = 20{,}340{,}000$. The total is hence 240,407,000 B.T.U.

The freezing-tubes, with 1·1 sq. ft. of surfacr per foot run or altogether $20 \times 150 \times 1{\cdot}1 = 3300$ sq. ft., can take up $3300 \times 85 = 279{,}500$ B.T.U. per hour. So that for the whole process $\frac{240{,}407{,}000}{279{,}500} =$ roughly, 850 hours, or over 35 days are required.

Such plants are always greatly dependent on the conditions, and in the daytime, and especially with long tubes, the losses due to radiation play a considerable part, so that arrangements should be made for an hourly supply of 350,000 B.T.U.

For the successful carrying out of the process, it is essential that the *separate freezing-tubes should be parallel.* If the deviations from parallelism are great, as may readily occur in deep bore-holes and may not be detectable immediately, the distance apart of the tubes in the watery soil becomes too great, so that the wall of ice formed is not sufficiently strong, and the success of the whole work—often lasting a month—is endangered.

Artificial Skating-rinks are prepared by freezing water placed in the bed of the rink to the height of from 2½ to 4 inches. The heat is abstracted either through the bottom, under which cool brine circulates in a flat, closed box with perpendicular dividing-walls (which carry part of the weight of the rink), or by means of a large number of pipes which are arranged in a gridiron formation all over the bottom, and which are traversed either by the brine or by the evaporating cooling-agent itself. In all cases the base must be not only somewhat elastic but also well insulated, in order to limit the absorption of heat from the exterior. When the attendance is large, the supports must be strong and are suitably constructed as arches carried by uprights. Fig. 189 is a sketch of the first method of cooling, which was devised by Linde and employed for the ice-rink at Nürnberg, while the second method, used for the "Palais de Glace," in Paris, is illustrated in Fig. 190. In the latter the tubes are arranged about $5\frac{1}{2}$ inches apart, one pipe receiving the brine from the entry pipe, while the adjacent one—connected with the first by a bent connecting piece—carries the brine back to the collecting pipe. This arrangement secures that almost all parts of the rink possess the same mean temperature. The two mains lie together in a side channel of the rink and when in use are likewise covered with ice.

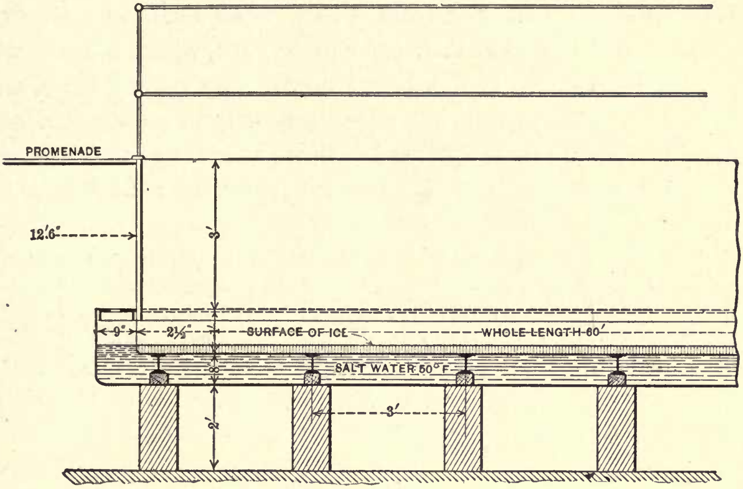

FIG. 189.

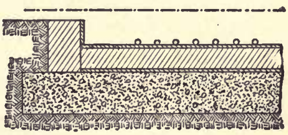

FIG. 190.

The cooling requirements of such a rink after freezing are limited to the removal of heat penetrating from outside, and this should hardly exceed 500 B.T.U. per square yard of the upper or under surface.[1] When the rink is much used, the wearing away of the ice surface becomes of importance, and the surface must then be planed and renewed by running on water. When the daily wear is from $\frac{2}{5}$ to $\frac{4}{5}$ of an inch, this requires, roughly, 350 to 650 B.T.U. per square yard, which must be supplied when the ice is not being used. The scanty experience gained up to the present indicates that the total cooling surface of the pipes should be from 1 to 1·5 times as great as the surface of the ice, the temperature of the brine varying from +10° to +14°.

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[1] Besides by direct conduction and radiation, the ice surface also receives heat from the heating and lighting of the room and from the freezing of the water-vapour in the air, which is increased by the large number of people present and which explains the cloud always found over the surface of artificial ice-rinks.

CHAPTER IX.

THE PRODUCTION AND APPLICATION OF COLD AT VERY LOW TEMPERATURES.

39. Methods of Pictet, Olszewski, and Dewar.—The preparation of a large number of substances (especially narcotics) in a chemically pure state is found to be greatly facilitated by crystallisation at temperatures between about −150° and −300°. Such abnormally low temperatures can, however, only be reached and maintained—as is necessary in these manufactures—by the evaporation of large quantities of gases such as *oxygen, nitrogen,* or *air* which have been previously liquefied at a somewhat higher temperature.

The liquefaction of these gases was first carried out in the year 1877 by the two physicists, Pictet and Cailletet, independently of one another, the means used being compression with the simultaneous removal of heat. Since the critical points of these bodies are very low, it became necessary to cool them somewhat below these temperatures; in Pictet's process a gradual lowering of the temperature is attained by the evaporation of increasingly volatile substances. Two bodies especially valuable for this purpose are *sulphur dioxide* and *carbon dioxide,*[1] the first of which was also successfully introduced by Pictet as a cooling-agent for compression machines. The use of *ammonia* for the same purpose had at that time just emerged from the experimental stage and had encountered a good deal of prejudice, which was, however, very soon overcome. Further, the critical

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[1] The critical data, boiling and solidifying points of these substances and of some other gases are contained in the table at top of next page.

TABLE XXIII.

	Critical Temperature.	Critical Pressure.	Boiling-point.	Solidifying-point.
Sulphur dioxide........	+313° F.	78·9 atmos.	+ 16° F.	−105° F.
Ammonia..............	+268	113 "	− 36	−107
Carbon dioxide.........	+ 88·45	72·9 "	−108	− 69
Nitrous oxide..........	+ 96·8	74 "	−112	−175
Ethylene..............	+ 50	ab. 55 "	−153	−272
Oxygen...............	−180	50 "	−296	−269
Nitrogen..............	−231	33 "	−317	−353
Air..................	−220	39 "	−312	
Hydrogen.............	−390	20 "	−406	

constants for ammonia (see table above) lie so near to those of sulphur dioxide that it is at least quite as good as the latter for the gradual attainment of low temperatures, the more so as its boiling-point at atmospheric pressure lies considerably below that of sulphur dioxide.

Pictet, whose method was until recently the only one by which difficultly condensable gases could be liquefied in large quantities, proceeded as follows. First of all, by means of *sulphur dioxide*[1] evaporating in a vacuum, *carbon dioxide* (later *nitrous oxide* was used) was liquefied, then cooled to about −150° F., and finally allowed to evaporate again under a low pressure while surrounding a tube filled with the gas, say oxygen (generated in some chemical reaction), by which means the temperature fell to about −220° F. (critical temperature of air). By continuous evolution of oxygen, the pressure of the latter could be raised to any extent, while the vapours of the carbon dioxide or nitrous oxide and also those of the sulphur dioxide, after taking up heat, were drawn away by compression pumps and condensed for further use. The liquefaction of the oxygen is a consequence of the simultaneous action of the high pressure and of the temperature falling below the critical point. In exactly the same way Pictet succeeded in converting other gases into the liquid condition; in more recent times he has extended his studies on the action of low temperatures, or, more accurately, the results of the withdrawal of heat at low temperatures, to a large number of other substances and organisms.

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[1] This always contained an admixture of carbon dioxide (Pictet's liquid), the action of which is of no importance. The process is described in the work of *Altschul*, "Mitteilungen aus dem Institute R. Pictet" (Zeitschrift für Kälte-Industrie, 1895), and also in that of E. *Meyer*, "Die Kälteerzeugungsmaschinen auf der Schweizerischen Landesaustellung in Genf" (ibid., 1897).

His methods for liquefying gases and also their applications

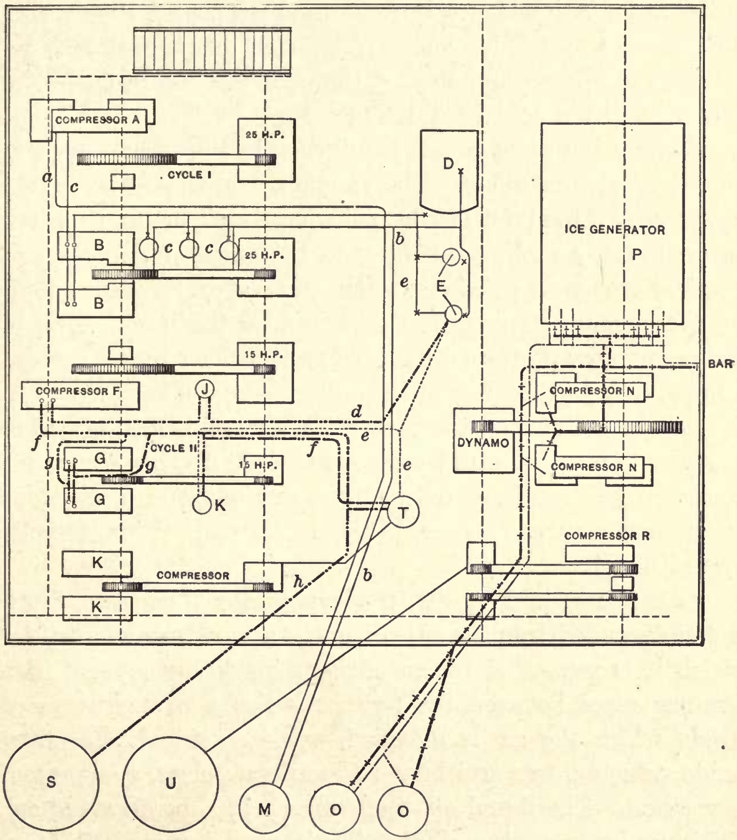

Fig. 191.

were demonstrated by *Pictet* at the Geneva Exhibition of 1896, the special pavilion employed being arranged as shown in Fig. 191. It contained, first, a sulphur dioxide compressor, *A*, which forced the vapours through the main *a* to the condenser *M* outside the building. By the pipe *b* the liquid sulphur dioxide was led to the evaporators *C*, *D*, and *E*, the last of which serves for the liquefaction of the nitrous oxide (at about −110° F.). In order to attain this low temperature the sulphur dioxide must be withdrawn at a very low pressure (about $\frac{2}{5}$ of an inch of mercury), which would not be possible with the valve-compressor *A* alone. Between the latter and the evaporators in question (of which *C* and *D* serve other purposes), Pictet, therefore, inserts two Burckhardt-Weiss disc air-pumps, *BB*, with pressure compensation, so that the condensation in cycle No. I is attained by a combined action.

Exactly the same course is followed in the second cycle traversed by nitrous oxide. The vapours, compressed in the main compressor *F* to about 140 lbs. per square inch, are first subjected to preliminary cooling by water in the cooler *J* and then pass into the condenser *E*, where, as already stated, they are liquefied by evaporating sulphur dioxide. Thence the liquid flows into the evaporator *J*, from which it is withdrawn by two vacuum pumps, *GG*, and led back to the main compressor *F*.

It is in the evaporator *T* that the air is liquefied after being brought to a pressure of from 2300 to 3500 lbs. per square inch, cooled in intermediate coolers by means of brine (cooled down in an ordinary cooling-machine), and dried. This completes cycle No. III.

As is shown in Fig. 192, the evaporator *T* consists, first, of a double-walled iron vessel, placed in a wooden cylinder, from which it is separated by an efficient insulating layer. In the annular space between the two walls lies a wrought-iron coil, inside which the air is liquefied, while outside it the nitrous oxide, which enters and leaves at various points, evaporates in a vacuum. The liquid air then collects in a bomb standing in the inner iron vessel, and from this it can be drawn off through a regulating-valve in cascades. The vapour here developed rises and surrounds the bomb, which can hence be uniformly maintained at a very low temperature.

The remaining apparatus of this very complicated plant calls for no special attention. All the compressors are driven by

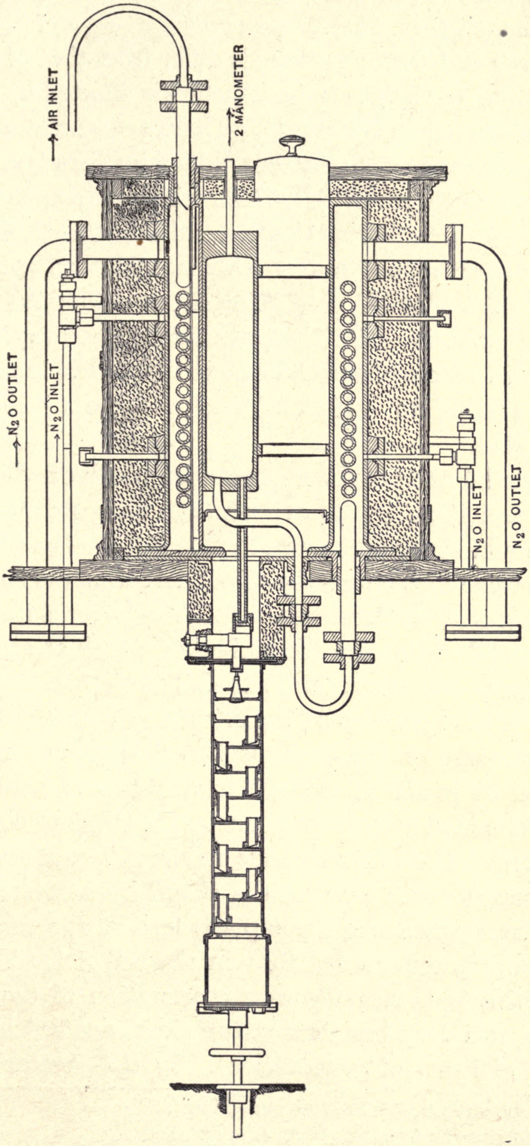
AIR INLET
2 MANOMETER
N_2O OUTLET
N_2O INLET
N_2O INLET
N_2O OUTLET

FIG. 192.

electric motors the powers of which are in some cases given in the above sketch (Fig. 191).

Olszewski's and *Dewar's* methods differ from that of Pictet in that the cooling of the gas to be liquefied is effected by ethylene (C_2H_4), evaporating in a vacuum. The vapour pressures of this substance were experimentally determined by Faraday, but only down to a temperature of −105° F., which corresponds with a pressure of 4·6 atmospheres. *Cailletet* and *Colardeau*,[1] *Olszewski*,[2] and *Wrobleswki*[3] found that the boiling-point at atmospheric pressure is −152·3° to −154·4° F., while, according to Olszewski, the freezing-point is −272° F.; so that by the evaporation of ethylene in a vacuum one would expect the critical temperatures of oxygen (−180° to −182° F.) and nitrogen (−231°) to be easily exceeded.

The essential parts of the apparatus in which ethylene is first condensed and then, by its evaporation, causes oxygen to assume the liquid state, are shown in Fig. 193.[4] The compressed gaseous ethylene first enters the spiral *AA*, which is situate in a strong-walled well-insulated vessel *B*. The latter is filled with a mixture of solid carbon dioxide and ether, which at atmospheric pressure causes the temperature to fall to −107° F.[5] By this means one is in a position to liquefy the ethylene in the spiral *AA* at a pressure of 4·8–5 atmospheres. By evaporating the cooling-mixture in *BB in vacuo*, a temperature of about −150° F. may be attained and the pressure on the ethylene correspondingly lowered. The ethylene thus rendered liquid enters at *C* a double-walled vessel *D*, which is closed by a rubber stopper and in which, by means of a pump attached at *J*, a considerably lower pressure is maintained than in the coil *AA*. The consequence of this is that a vigorous evaporation of the ethylene takes place at a very low temperature, accompanied by a rapid absorption of heat (of evaporation). In the absence of other sources, this heat is withdrawn from the oxygen which circulates in the coil *OO*, and which is delivered from a preparation appa-

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[1] Comptes Rendus, 1888, **106**, page 1489.

[2] Wiedemann's Annalen, 1889, **37**, page 337.

[3] Sitzungsberichte der Wiener Akademie, 1888, **97**, page 1378.

[4] From the Philosophical Magazine, 1895, page 301.

[5] *Cailletet* and *Colardeau*, Comptes Rendus, 1888, **106**, p. 1631.

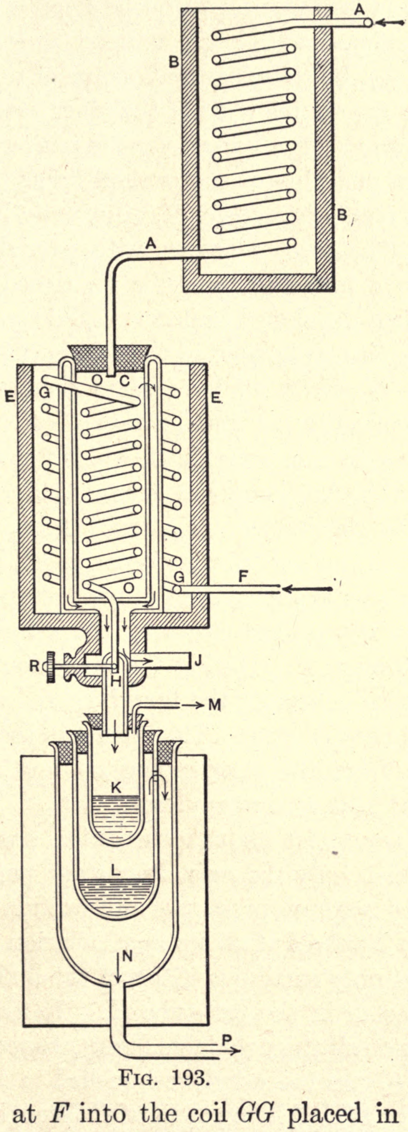

FIG. 193.

ratus or pump at *F* into the coil *GG* placed in the vessel *EE*, where it undergoes a considerable preliminary cooling by a mixture of carbon dioxide and ether. The pressure of the oxygen varies from 20 to 50 atmospheres. Since the cooling-mixture in *EE* has an appreciably higher temperature than the ethylene evaporating in the vessel *D*, heat might be given up to the latter. This is, however, almost completely avoided, at least during the formation of vapour, by drawing the vapours of the evaporating ethylene between the double walls of the inner vessel *D*. In the channel through which the ethylene vapour passes to its exit, *J*, is placed also the exit-pipe of the oxygen liquefied in *OO*, this pipe ending in a regulating-valve, *H*, adjustable by means of a hand-wheel, *R*. The oxygen passes, with partial evaporation, through this valve into the test-tube *K*, closed by a rubber stopper; the vapour developed while the liquid runs through this valve—which is quite analogous to the regulating-valves of the cooling-machine—is drawn off by a glass tube at *M*, the remaining liquid collecting at the bottom of *K*. In order to minimise the entrance of external heat, which must be very considerable owing to the extremely low temperature of the contents of *K*, the latter is surrounded by a wider tube, *L*, containing liquid ethylene. This outer tube is also arranged inside another one, *N*, and into this the insulating ethylene-vapour passes by an aperture near the rubber stopper; the ethylene is finally pumped away through *P*. During the operation the whole of the arrangement for collecting the liquid oxygen is carefully insulated—best by sheep's wool. This prevents the precipitation of water-vapour from the surrounding air and the relatively great evolution of heat accompanying it.

It is seen from the above description that the whole apparatus, of which only the most important parts are shown in the figure (the various pressure- and suction-pumps being omitted), is very involved; that its manipulation must be very difficult, and that only small quantities of the body to be liquefied can be collected in the test-tube *K*. It must further be mentioned that not all the avoidable losses are removed. First of all, in order to use the ethylene liquefied in *AA* more efficiently, it would be advisable to pass it at *C* through a regulating-valve like that at *H*. Besides this, it is a mistake to carry the exit-tube of the liquid oxygen through the channel by which the ethelyne is pumped away and also to place the regulating-valve *H* in it, since, in traversing the double walls of *D*, the ethylene has an opportunity of becoming warmed by the warmer cooling-mixture in the vessel *EE* and of unnecessarily transferring a part of this heat to the oxygen in the exit tube. In spite of the shortness of the latter, this transference of heat may be quite considerable, since the conducting power of metals for heat, like that for electricity, rapidly increases as the temperature falls. This loss can, however, be greatly lessened by surrounding the oxygen exit tube as far as *H* with evaporating liquid ethylene, thus connecting the space in question with the interior of the vessel *D*; the ethylene-vapour would then be drawn off at another place.

It must further be pointed out that the process just described does not differ essentially from that of *Pictet,* since the temperature is lowered by stages by the evaporation of substances having lower and lower boiling-points.

To ascertain the efficiency of this method, we shall determine the *expenditure of work necessary* to liquefy 1000 lbs. of oxygen per hour at −261° F. The heat of evaporation of this substance was found by *Dewar* to be 144 B.T.U. per lb. at atmospheric pressure, and for the above temperature, corresponding with a pressure of 22 atmospheres, roughly 110 B.T.U. To this must be added the cooling from +68° to −261° F., which, with a specific heat[1] of approximately 0·27, requires about 90 B.T.U.; the effective cold required is hence 200 B.T.U. per lb., or 200,000 B.T.U. per 1000 lbs.

Let us now assume, without treating in detail the cooling-agents used (which, as seen above, may be very varied), that the process takes place in three steps, in the first the heat taken up at −261° F. being given out at −148°, from which it is brought to −4° and then finally absorbed by cooling-water at about 50° F.; we can then ascertain the minimum quantities of work necessary for the separate processes. In consequence of the great increase of the radiation of heat from outside—in spite of the best possible insulation—we must assume that at $-261°$ F. at least double the cooling effect made use of, i.e. 400,000 B.T.U., has to be supplied.

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[1] According to investigations by *Lusanna* (Nuovo Cimento, 36, page 70), the specific heat of gases increases very considerably with the pressure.

In order to take up this amount of heat and give it out again at $-148°$, the power required in a cooling-machine having an efficiency of 0·6 is about 150 indicated horse-power,[1] of which the heat-equivalent to the extent of 380,000 B.T.U. is added to the original amount in the second stage, so that here at least 780,000 B.T.U. have to be overcome. Besides this there are losses by radiation, and in this stage these must be taken as at least 360,000 B.T.U., so that the real amount of heat to be removed is raised to 1,140,000 B.T.U.

To remove this between the limits $-146°$ and $-4°$ with a machine having an efficiency of 0·60, 350 indicated horse-power with an hourly heat-equivalent of 900,000 B.T.U. must be applied so that in the third step 2,040,000 B.T.U. have to be overcome. Taking the loss by radiation at its normal value, namely, about 12 per cent, this amount is raised to, roughly, 2,300,000 B.T.U., so that about 100 I.H.P. are required. The total power is hence $150+350+100=600$ I.H.P., the yield being about *1·6 lbs. of liquid oxygen per I.H.P.*[2] This does not include the energy used in the preparation of the oxygen from chemical compounds.

40. Linde's Method.—The problem of liquefying difficultly coercible gases, especially atmospheric air, has been recently attacked by Professor *C. Linde* of Munich and solved by him in a very simple way, quite different from any previously described. Linde's method is based on deviations from the laws of Mariotte and Gay-Lussac, which, as is well known, are not perfectly obeyed by any gas. These deviations depend on the reciprocal attraction exerted between the separate molecules and therefore become greater as the density of the gas increases, i.e., as its pressure rises and its temperature falls. If a gas which—apart from these attractive forces—follows exactly the laws of Mariotte and Gay-Lussac is allowed to flow from an aperture under pressure, its temperature can only become lowered owing to the presence of unequalised energy of motion. If it comes to rest again, this fall of temperature—which can only be small—vanishes. This requirement of theory was experimentally tested in 1862 by *Thomson* and *Joule*, who obtained considerable lowering of the temperature[1] both for carbon dioxide and for dry air.

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[1] The calculation is made on the basis of Carnot's cycle, by the method explained in § 4.

[2] By distributing the lowering of temperature over the three steps, the yield may be raised to, roughly, 2 lbs. per I.H.P.

Starting from these researches, Linde proceeded to construct an apparatus by which it is possible to liquefy atmospheric air without using any substance (such as ethylene, etc.) other than cold water for absorbing the heat of compression. The greatest difficulty was doubtless the necessary initial lowering of the temperature of the space in which the liquefaction was to take place, that is, in the attainment of persistent, unchanging conditions. Without making use of the evaporation of readily liquefiable gases, the air may be lowered in temperature by passing it through a throttle-valve, and air thus cooled may

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[1] The amounts of cooling are given in the following table for various initial temperatures and for a difference of pressure of 1 atmosphere on the two sides of the throttle-valve.

TABLE XXIV.

Initial Temperature.	Fall of Temperature.	
	For Air.	For Carbon Dioxide.
°F.	°F.	°F.
32	0·497	2·684
44·8	0·473	2·487
96·1	.....	1·836
102·4	0·403	
129·2	.....	1·589
199·0	0·274	
200·3	.....	1·161
207·5	.....	1·152

be employed in an economiser to cool the air before it reaches the valve. By repeating this process the temperatures before and behind the throttle-valve become continually lower until finally the condensation-point is reached. The economiser used by Linde consists of a coil of two pipes, one of which, as shown in Fig. 194, passes inside the other; these pipes are about 100 yards long and have bores of about 1¼ and 2 inches respectively. Since all that is desired is a passage of heat from the contents of the small, inner tube to the surrounding air in the wider tube, the whole spiral must be carefully insulated from the outside. This can be effectively done by means of sheep's wool. The double spiral, the separate coils of which do not, of course, touch, but are also insulated from one another by sheep's wool, is fixed in a tall, cylindrical wooden vessel; the wider tube enters at *F* the iron pot *T*, into which also the narrow tube just after leaving the outer one at *E* and being provided with a regulating-valve *R* adjustable from outside.[1]

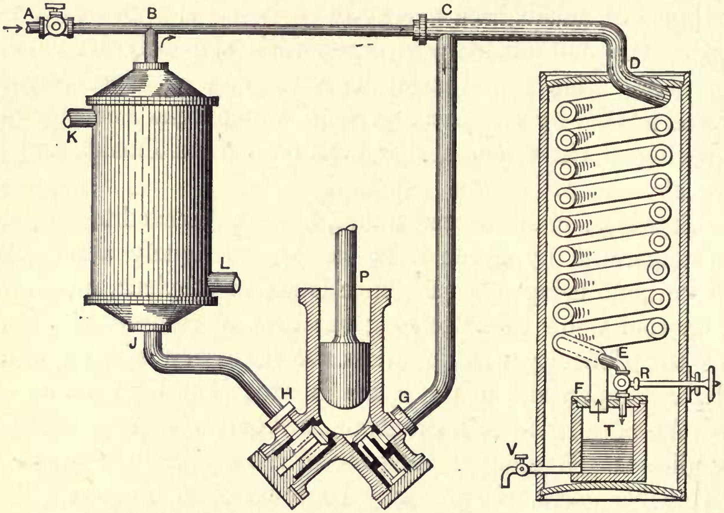

Fig. 194.

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[1] In the first experimental installation of the *Linde Ice-Machine Company* in Munich, the economiser, together with the covering and collecting vessel, had a weight of, roughly, 1¼ tons.

While the temperature is being lowered, the air-content of the apparatus proceeds through one cycle after another, the whole of the cooled air passing through the throttle-valve *R*, rising into the outer tube at *F* and cooling the air in the inner spiral nearly down to its own temperature. Outside the economiser, the two tubes separate at *C*, the wider one leading at *G* to the suction-valve of a pressure-pump *P*, while the narrow pressure-tube is connected with the pressure-valve *H* of the same pump. Further, in this pressure-tube is inserted a cooling-apparatus *J* to carry off the heat of compression. *K* and *L* are the inlet and outlet for the cooling-liquid, which is usually ordinary well-water. The construction of the cooler is quite similar to that of the condensers used for cooling-machines which have been already described. If this is arranged on the counter-current principle, the air can be easily cooled nearly to the initial temperature of the water. About this same temperature is possessed by the air drawn from the apparatus *D* by the pump *P* (which hence works under normal temperature conditions), owing to the intense absorption of heat from the air between *F* and *C*.

Since, according to the experiments of Thomson and Joule, the fall of temperature during the passage through the valve *R* is nearly proportional to the fall of pressure, it is advisable, in order that the critical point of the gas may be quickly reached, to make the higher pressure in the machine as great as possible —about 65 atmospheres. If then the air is allowed to expand again to atmospheric pressure, a very considerablel owering of temperature would take place in the compressor *P*. On this account the suction pressure is also chosen fairly high, namely, about 22 atmospheres (slightly less than that in carbon dioxide cooling-machines); a further advantage of this is that the amount of air circulating is increased and the arrangement more efficiently used. Moreover, with a higher pressure, it is not necessary for the temperature in the condensation vessel to fall so low before liquefaction begins. When the dimensions of the apparatus are large, the lowering of temperature requires, however, a considerable time, in some cases 17 hours;[1] this is readily comprehensible in view of the unavoidable losses caused by penetration of external heat and the necessity for cooling the large masses of metal contained in the double spiral and the vessel *T*.

This penetration of heat naturally increases as the temperature in the interior of the spiral and in the vessel *T* falls and reaches its maximum at the moment when liquefaction begins. The pressure will then necessarily sink rapidly in all parts of the apparatus, since the liquefied air no longer circulates. Hence, in order to render continuous working under constant conditions possible, it is necessary to replace the liquid in *T* by pumping in fresh air. This is effected by a compressor (not shown in the sketch), which draws in fresh air rendered almost perfectly dry by passage through a cylinder filled with calcium chloride and compresses it to the maximum pressure in the liquefaction apparatus. The heat of compression is previously removed from this air in an apparatus similar to the cooler *J*, so that when it enters at *A* it can immediately unite at *B* with the air already in circulation.

The drying of the new supply of air must be carried out with the greatest care, since not only would the condensation and subsequent freezing of the water-vapour liberate considerable quantities of heat, but the ice formed would quickly block the pipes and render further working impossible. These difficulties were completely overcome in Linde's experimental apparatus, so that no disturbance was caused by condensation of water or formation of ice.

Finally, the liquid formed can be drawn off by the valve *V* placed outside the cooler. In general, it exhibits the properties of liquid oxygen, of which it contains up to 70 per cent, while atmospheric air only contains some 20 per cent. It was also found, as had been observed by Dewar, that the nitrogen present in the liquid evaporates more rapidly into the air than the oxygen; after a time the originally colourless liquid which boils vigorously in the air, becomes bluish, this being the colour of liquid oxygen.

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[1] This time and also the ratio of the losses by radiation to the useful yield are diminished by increasing the amount of air passing through the apparatus, i.e., by using larger or quicker-acting compressors.

The principle of Linde's apparatus is also employed in an arrangement which has been recently constructed by W. *Hampson* of London, and which, on account of its compactness, is especially suited to use in the laboratory. It only differs from that of Linde in that the passage to the regulating-valve does not take place in a narrow tube arranged inside the return tube, but is effected by a thin copper tube with numerous coils placed in the inner space of the apparatus filled with the return gases. From the rapid action of this arrangement, with which liquid air may be obtained in 16 minutes, it is seen that more depends on the size than on the form of the surface through which the heat is withdrawn. Further, with Hampson's apparatus, the compressor may be omitted and the air fed in from a cylinder under about 200 atmospheres pressure.

Such pressures had already been used by Linde in the working of his apparatus for laboratory use shown in Fig. 195.

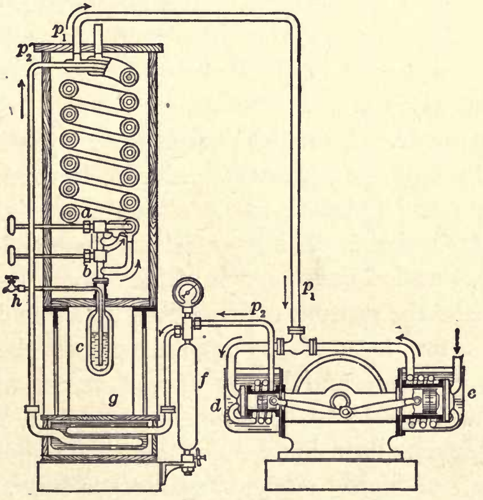

FIG. 195.

He started here from the fact (based on the laws of gases) that the work of compressing a given quantity of a gas is determined not by the absolute pressures, but by the ratio of these pressures, so that, for example, the compression of 1 lb. of air from 100 to 200 atmospheres pressure requires the same amount of work as compressing it from 1 to 2 atmospheres. Since, on the other hand, the Thomson cooling effect increases with the difference of pressure before and behind the regulating-valve, it is to be expected that, by using higher pressures, a more rapid and intense cooling would be obtained with a relatively smaller expenditure of work than would be the case with more moderate pressures.

Linde's apparatus hence works with two compressors, of which the larger one, *e*, takes atmospheric air and pumps it at a pressure, p_1, of 16 atmospheres into the suction-pipe of the smaller cylinder, *d*, which then raises the pressure, p_2, to 200 atmospheres. The cylinders are provided with water-jackets, in which are also placed coils for cooling the compressed air. The high-pressure air now passes through a bottle, *f*, which is furnished with a manometer and in which the moisture and lubricating-oil are separated; after subsequent cooling in *g* in the innermost tube of the triple spiral of the liquefaction apparatus, it is throttled through the regulating-valve *a* down to 16 atmospheres and then, for the most part, flows through the middle tube back to the suction-tube of the high-pressure compressor *d*, where it is mixed with the air coming from the low-pressure compressor. A smaller part of the air passing through *a* proceeds through the regulating-valve *b*, beyond which the lowest temperature is attained at a pressure slightly above that of the atmosphere. The liquefied air then collects in the Dewar's double-walled vessel *c*, from which it can be run off by the cock *h*, while the gaseous residue gives up its cold in passing through the other tube of the apparatus into the open air. With such a compact and handy apparatus, in which liquefaction begins from 1½ to 2 hours after starting, the yield—after the stable condition has been attained—is about 1¾ pints (about 2½ lbs.) of highly oxygenated air per hour, the power used being about 3 H.P., i.e. 1 H.P. for every 0·8 lb. of air per hour.

The *industrial application* of such liquefied gases, and especially of air, is greatly hindered by the impossibility of storing them for any length of time. Liquid air (termed *oxyliquid*), however, is not only of value on account of its intense cooling action but has also been introduced into explosives. Cartridges (up to 2½ inches diameter) filled with wood charcoal are now prepared and are saturated with liquid air close to the place where they are to be used; they retain their full explosive power for at least 15 minutes. The insulation of the cartridges is effected by means of paper coverings and the ignition in the ordinary way by a fulminating mercury cap and Bickford string.

The importance of Linde's process lies less in this, however, than in the possibility it gives of *separating the two constituent gases of the air by mechanical means.* To this end, Linde liquefies (see Fig. 196) the highly compressed air in the pressure-tube in front of the regulating-valve r_1 by passing it through a spiral placed in the collecting vessel *S*; the highly oxygenated liquid in the latter is passed through a further regulating-valve r_2 into a second counter-current apparatus, in which it completely evaporates and so imparts its low temperature to the air entering this apparatus, which also consists of two tubular spirals one within the other. The highly compressed air is previously led into two pressure-tubes, *O* and *N*, which only join immediately before the collecting vessel; if, now, the pressures are so regulated that in the collecting vessel there prevails only a small excess of pressure over that outside, it is possible, while recovering the whole of the cold in the liquefied air, to obtain almost pure nitrogen at the extreme end of the one counter-current apparatus and, at the other, oxygen at atmospheric pressure and 32° F.

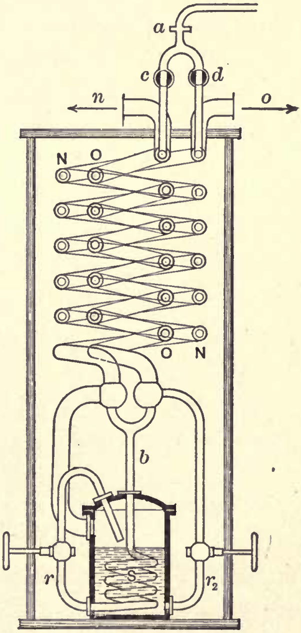

FIG. 196.

Concerning the yield of oxygen—which naturally cannot be pure—obtained in this way, no experimental data have as yet been published, so that it would be premature to give an opinion as to whether it would be possible for this method to compete with the chemical methods already in use (e.g., Brin's barium peroxide process, by means of which oxygen absorbed from the air can be regenerated).

Even if all the expectations of Linde's method should not be realised, it is, however, certain that the technics of low temperatures has been enriched by a new method, which is not only more practical, but far simpler, than the older processes and which is worthy of a place alongside the important developments already effected by Linde in ammonia compression machines.

CHAPTER X.

DETERMINATION OF THE YIELD OF COOLING-MACHINES.

41. Yield of Cold from the Ice-production.—The continually increasing competition in the refrigerating industry has been the cause both of various practical evils—mainly in the conditions of payment—and of considerable progress in the construction of cooling-machines. This is shown more especially in the precision of the guarantees offered—and generally maintained—with reference to the yield and work or steam consumption of these installations. More stress is now laid on these guarantees, which were formerly but seldom given, and, in spite of the expense, such investigations are often put into the hands of impartial experts. The methods made use of will be here shortly described and the results they give criticised; this is rendered all the more necessary in the case where one has, from the experimental data, to draw conclusions concerning the practical working, which often cannot be stopped even for a short time, although by not doing so the accuracy of the results obtained may suffer. However, with care an experienced man can nearly always attain sufficient accuracy to allow of the most important of the data guaranteed being tested.

We need not here go into the question of the *compressor work*, which has been already dealt with in detail in § 20.

The tests made are generally known as *calorimetric tests,* since their main object is to determine a quantity of heat, namely, the yield of cold per unit of time under definite conditions of temperature. Quantities of heat are, however, only detectable by means of the bodies with which they are associated, and the nature of these bodies, especially their state of aggregation, has to be taken into account in these measurements.

Apparently the most convenient measurement is the *weighing of solid bodies;* the cold generated by a machine may, for example, be determined from the weight of ice formed, the heat of fusion of the latter being known (142·2 B.T.U. per lb. at atmospheric pressure) and practically invariable. Although, however, there is no difficulty in measuring the temperature of the water before its introduction into the ice-cells of the generator, yet the temperature of the ice formed—mostly the mean temperature of the brine in the generator, when the freezing occupies a long time—is less certain; besides which it must not be overlooked that only a few blocks can be weighed, since the complete weighing of whole series of cells, both full and empty, would occupy too much time. But the removal of the blocks is only possible after the cells have been dipped into a thawing vessel, so that there is a certain loss in every weighing, since it is impracticable to collect the water when the block is tipped out. These difficulties are generally obviated by weighing a few blocks as tests —care being taken that the various series of cells are filled as nearly as possible to the same extent—and by estimating the loss due to melting at from 5 to 10 per cent. Even with the most careful determinations of the temperature, the final result given by this method cannot be very exact, so that the *measurement of the cold by means of the ice-production* must be regarded as the *most inaccurate* experimental method. It may, however, be improved by filling the cells as uniformly as possible with water and determining the amount of the latter by measurement of the filling vessel. When the latter is irregular in shape, the best way is to place in it a previously weighed quantity of water sufficient to fill all the cells of a series and to mark the height of this water in the vessel. This same quantity of water is then used for every series. If the filling vessel is furnished with an *overflow* arrangement, the lower edge of this may conveniently be used as the height to which the vessel is to be filled. No notice need then be taken either of the extent to which each separate cell is filled or of the melting losses occurring when the ice is removed. This method is certainly far more accurate than taking separate cells, the mean content of which is determined by the height of the water. It should, however, not be forgotten that, in view of the uncertainty accompanying the measurement of the real mean temperature of the ice blocks, the result is still attended with an inaccuracy which can never be quite removed but can be diminished by allowing the finished blocks to remain for some time in the generator. In this way the temperature inside the block becomes the same as that of the outer surface, but this takes place slowly owing to the small heat-conductivity of the ice. Since the specific heat of ice is about 0·5 and the mean temperature of the brine in most generators is approximately 21° F., the cooling of the ice from the freezing-point to the brine temperature corresponds with a heat-quantity of $0\cdot5\ (32-21)=5\cdot5$ B.T.U. per pound. If each of the blocks is removed immediately after the freezing of the inner portion, which cannot then have a temperature lower than 32° F., the mean temperature of the ice may be taken as 27° or 25°, corresponding with the abstraction of 2·5 or 3·5 B.T.U. per pound during the cooling. The errors in this method will be about equal to this difference of say 3 B.T.U. per pound. If the ice is made from water at 50° F., to freeze 1 lb. will require the removal of $(50-32)+142=160$ B.T.U., or with the cooling of the ice, 165·5 or 163 B.T.U., so that the error amounts to somewhat less than 2 per cent. This may be regarded as satisfactory, since, as we shall see later other methods do not admit of greater accuracy than this.

Although the determination of the yield of an ice-machine is improved by measuring the amount of water frozen instead of the ice formed, yet this only gives the *net yield* of the machine. If this is compared with the corresponding value for a refrigerating-machine to which heat is led by circulating brine (as, for example, in the cooling of brewery cellars), it will generally be found to be greater in the latter case—the work done by the engine and also the evaporator and condenser temperatures being the same in the two instances. The explanation of this is then obtained from the balancing of the heat, i.e., the simultaneous determination of the indicated compressor work and the condenser yield. On subtracting the equivalent of the former (i.e., the H.P. multiplied by 2545, the number of heat-units generated by the destruction of 1 H.P.) from the latter, the *gross yield* of the machine is obtained; with ice-machines this generally comes out considerably greater than the above net yield. If now the condenser works at a temperature differing but little from that of the surrounding air, the influence of external conditions on it may be neglected; the difference between the gross and net yields then represents the *losses* caused by the conduction and radiation of external heat into the generator. On account of the large space occupied by the ice-cells and the surrounding brine, the surface exposed to these disturbing influences is much greater in ice-generators than in ordinary evaporators; the relatively considerable losses occurring in these generators is thus easily explained.

Finally, it may be pointed out that with the process described above, reliable results are only obtainable when the conditions are stable; that is, when the temperature of the brine is maintained by replacing the ice removed by fresh water regularly and as far as possible in the same intervals of time. In order to avoid all more or less inaccurate corrections, the final and initial temperatures of the brine should be identical. If the brine temperature rises towards the end of the experiment, the removal of the ice should be delayed, whilst if the temperature falls the removal should be hastened. It must, however, be made quite certain that the cells are frozen through, since otherwise a very considerable error may arise.

42. The Cooling Method.—In some cases the stable state of the cooling-machine cannot be obtained, either in consequence of occasional insertion of fresh-water coolers and ice-generators in the brewery, or because the persons carrying out the test have had no time at their disposal for the preliminary preparation. Under these circumstances there is generally nothing left to be done but to exclude all external sources of heat for a short time and to cool the evaporator alone with its brine from a definite higher temperature to a lower one. If the evaporator serves also as an ice-generator the cells are to be removed and so much brine added that the latter stands at its normal height. If, then, the stirring-apparatus remains in action, the machine works—apart from the steady fall of temperature in the evaporator—under normal conditions. It need hardly be emphasised that the brine temperature must be read off, not only at short intervals—say every ten minutes—but also at several positions in the generator, so that reliable average values may be obtained. The difficulty of determining the brine temperature at great depths may be easily overcome by fastening the thermometers in tin boxes (in an emergency tins used for preserves may be used), as shown in Fig. 197, by means of a long wooden rod and lowering them into the apparatus by means of leaden weights. When rapidly removed the thermometer remains in the same brine which surrounded it in the depths of the liquid, so that it gives the temperature without any appreciable error. If, however, as in the Linde apparatus, there are side-tubes jutting into the brine and filled with glycerine or brine, the bulb of the thermometer should be surrounded with cotton waste, etc., so that when the thermometer is removed for the purpose of reading, the thread shall not rise owing to the influence of the external temperature.

After these few preliminaries are arranged, the test—in which, when possible, indicator diagrams of the compressor should be taken and the efficiency of the condenser (to which we shall return later) determined—proceeds quite smoothly and, when the initial temperature of the brine is 32° and the cooling carried to 20°–18°, is finished in from 1 to 2 hours.

The mean observed temperatures are conveniently shown graphically as a function of the time, a diagram like that shown in Fig. 198 being so obtained. If the revolutions of the engine

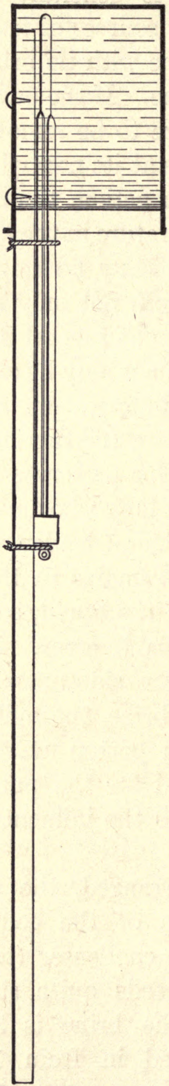

FIG. 197.

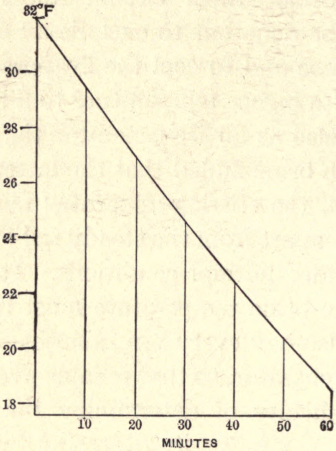

FIG. 198.

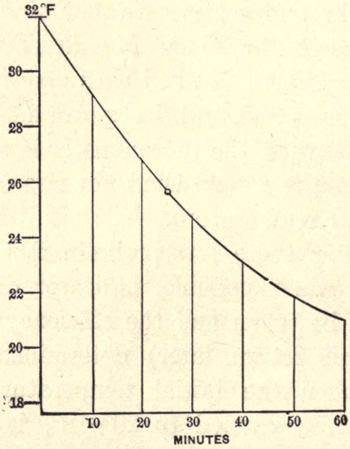

FIG. 199.

undergo no considerable fluctuations during the experiment, and if also no disturbing influences come into play, the temperature curve will have a steady course, the temperature falling at first rapidly and later on more slowly. The continuous nature of the curve allows any erroneous readings to be subsequently corrected, so that the drawing of the temperature curve should never be omitted. This curve is, besides, often of great use in calling attention to a disturbing factor in the test. If, for instance, the temperature curve exhibits a striking irregularity at any point, as, for example, at 25·5° F. in Fig. 199, it must be concluded that a previously inactive source of heat has suddenly come into action. According to the author's experience, this is always due to the separation of salt from the too dilute brine on the surface of the evaporator-tubes.

The correctness of this assumption can be easily tested by comparing the temperature inside these tubes, as indicated by the manometer of the evaporator, with that prevailing at the same time in the brine. Under the circumstances, differences of from 15° to 18 ° F. will always be found, whilst under normal conditions where the tubes are not covered with an insulating layer of ice and salt, this difference only amounts to from 7° to 11° F. This phenomenon—which is especially frequent with air-cooling apparatus where the air to be cooled comes into contact with the brine and where the brine diluted in this way has not been strengthened by the addition of salt—indicates, therefore, a very uneconomical working of the plant. It will be seen from this that it is not only necessary to make the salt-content of the brine sufficiently high to prevent the lowest temperature it attains from freezing it, but that this latter should only occur at a temperature near to that prevailing in the interior of the tubes.

In order to ascertain the influence of the surroundings on the apparatus, the cooling experiment is often followed by a *heating test*, in which the compressor is shut off and the regulating-valve closed, but the stirring arrangement kept working; if possible, this test should be continued until the original temperature is reached. This action is slower the nearer the external temperature—supposed constant—lies to that of the brine and the better the apparatus is insulated. The temperature curve corresponding with this experiment will have the appearance shown in Fig. 200. If, however, the tubes are coated with separated salt or ice, this is first of all dissolved under the influence of the heat penetrating from outside, so that there is often no rise of temperature observed for a long time. In this case there is no object in the heating experiment, as from the preceding cooling experiment the temperature curve of Fig. 199 has been found to be of no use, at least below the position where the irregularity occurs.

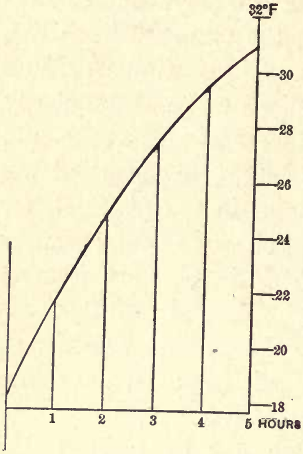

FIG. 200

Concerning the *application of these experimental results in determining the cooling effect,* it is usual, first of all, to calculate the water-equivalent, *W,* of the cooled mass from the amount of brine in the evaporator and the weight of metal (generally obtainable from the maker or from the drawings of the machine) in the vessel and the tubes in it, together with the specific heats of the brine and metal. No notice is here taken of the insulation, the change of temperature of which cannot be observed, while its weight and specific heat are never accurately known; but, on the other hand, the cooling of the metal parts is taken as the same as that of the brine and is hence undoubtedly too high. This neglect of the insulation and the assumption regarding the cooling of the metal parts constitute the weakness of the whole process, and this cannot be removed by a heating test, which gives the external heat with the same inaccuracy. A certain amount of control is rendered possible by the above-mentioned simultaneous determination of the compressor and condenser yields, from which the theoretical cooling effect, Q, can be obtained. If, for instance, the cooling per hour over the interval of temperature given in the maker's guarantee is $t°$, while, in a heating experiment where the mean temperatures of the brine and surrounding air are as before, the former rises by Δt per hour, the following equation must be fulfilled:

$$Q = W(t + \Delta t). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

From this W can be calculated and the value obtained compared with that derived from the weights and specific heats of the brine and vessel. We may, in general, be satisfied with agreement within 4 per cent, and can then take the *magnitude* Wt as the *net cooling effect*. Although this method is not very accurate,[1] it is frequently employed, and not only on the grounds stated at the outset but also because it permits of the determination of the relative diminution of the cooling effect—other conditions being the same—as the evaporator temperature falls. I have often used it successfully for this purpose and have obtained from it valuable information on the behaviour of machines as the temperature falls.

The error introduced by the inaccuracy of the water-equivalent W—which is especially serious when, with air-condensers, the gross cooling effect cannot be ascertained and the value of Q, hence, not controlled—can be removed by a second test in which W is increased by a known amount. To this end the brine contained in the apparatus is simply increased or diminished in amount; if, then, w is the water-equivalent of this addition, we have, for the same cooling effect,

$$Q = (W + w)(t' + \Delta t'). \quad . \quad . \quad . \quad . \quad . \quad (2)$$

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[1] The water-equivalent of the metal portions of the vessel should of course not be neglected, as, especially with medium-sized and small evaporators, this may easily amount to 20 per cent of that of the brine.

From (1) and (2) it follows that

$$W=\frac{w(t'-\Delta t')}{t-t'+\Delta t-\Delta t'}. \qquad \ldots \ldots \quad (3)$$

This method has the great advantage that the whole measurement of the water-equivalent is made on a body of relatively small weight outside of the evaporator and can, therefore, be carried out accurately. The latter is all the more necessary, because, as is evident from equation (3), all errors here made bear the same ratio to the magnitude of W. If, for example, the first test gives a fall in temperature, $t=6\cdot3°$ F., and a subsequent rise, $\Delta t=0\cdot36°$ per hour for the same mean temperature, then

$$Q=W\times 6\cdot66.$$

After adding 1000 lbs. of brine with the specific heat 0·9 (corresponding with that already present), the water-equivalent of which is hence 900 lbs., a second test gives, for the same cooling effect, $t'=5\cdot8°$ and for the heating up 0·36°, as before; hence

$$Q=(W+900)6\cdot16.$$

Equating these two expressions for Q, we obtain

$$W=\frac{900\times 6\cdot16}{6\cdot66-6\cdot16}=11{,}088.$$

The accuracy here depends not only on that with which W is determined, but also on the measurement of the temperature, with which the greatest care must be taken. For this the temperature diagram, explained above, is of great value. Finally it may be mentioned that water alone should not be added to the generator to alter the value of W, since this might easily lead to a brine which is too weak and might cause disturbances such as are indicated in Fig. 199.

43. Determination of the Cooling Effect at Constant Temperature by the Condensation of Water-vapour.—If the guaranteed cooling effect refers to a constant temperature (say 23° F.) in the brine-bath, the cooling method allows us to determine, with moderate accuracy, the yield over a temperature-interval comprising the specified temperature as the mean value, but only where it is possible or admissible to add or remove a certain quantity of solution so as to arrive at the true water-equivalent of the whole mass. This is, however, scarcely ever the case, especially with air-cooling apparatus, and the results obtained by the cooling method can often only be taken as rough approximations. General approval should, therefore, be accorded to *Pressl's* suggestion to compensate the cold developed by the condensation of water-vapour and to weigh the condensed water, especially as the actual net cold generated is determined independently of the losses by radiation and conduction, and the time required to obtain reliable results is small. The author has made repeated use of this method in recent years and has obtained very satisfactory results. The complete or partial annulling, by water-vapour, of the cold developed has been previously used experimentally, but the measurement of the condensed liquid is new. This measurement calls for a number of precautions, neglect of which may easily spoil the whole test. We shall first give a sketch of the method itself.

In the apparatus to be tested (air-cooler, etc.) is placed, perpendicular to the direction of flow of the brine, a coil made of thin-walled but rigid copper tubing[1] (see Figs. 201 and 202), the two ends being above the brine-bath and provided with ordinary flanges. To one end is attached first a manometer and then a steam-valve, just above which is a water-trap, which is to be so arranged that during the test it allows not only all the water from the steam-pipe, but also a strong current of steam, to pass into the air. Only in this way can it be certain that what enters the coil is really dry steam, the temperature of which is given by the manometer reading. In order to maintain a uniform pressure in the coil during the test, the other end of the coil is provided with a valve the aperture of which is sufficiently great to allow the water formed to escape under the pressure inside. This is then conducted by means of flexible hose to the measuring vessel, which should be placed as near as possible and preferably on the scale-pan of a weighing-machine. During the test the two valves on the coil are so regulated that, on the one hand, the outflow of water is as nearly as possible uniform and, on the other, the temperature of the brine remains practically constant. For the test to succeed, these matters must have the continuous attention of a competent fitter or engineer.

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[1] The coil is to be submitted to the same test-pressure as the boiler which generates the steam and must also be tested under water to see if it is airtight.

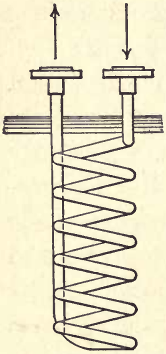

FIG. 201.

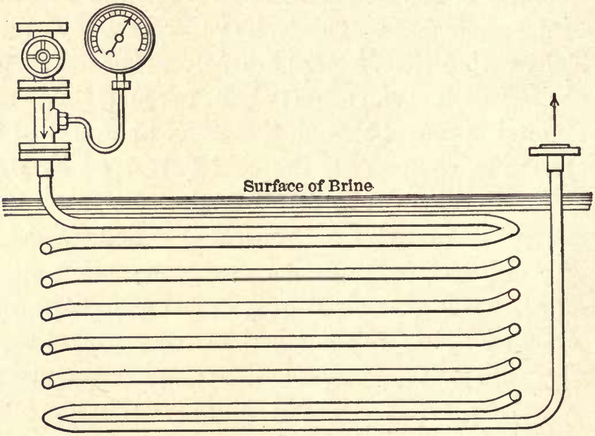

FIG. 202.

Before turning to the description and use of the data obtained, I may make a few remarks on the dimensions of the coil in relation to the cooling effect expected. First of all, the formation of ice at any place inside the coil—which would rapidly cause it to become frozen up—must naturally be avoided. This requires that the condensed water shall not only flow but a short distance in the coil, but that it must have at least a certain minimum velocity, which I have ascertained to be about $4\frac{3}{4}$

inches per second. But this velocity determines, for any given (or rather, expected) cold evolution, the maximum diameter for the tubing of the coil, while the length of the latter is given by the amount of heat passing per square foot of mean heating surface per hour. If the excess steam pressure in the coil is kept at about 42·5 lbs. per square inch (corresponding to a temperature of, roughly, 290° F.), while the temperature in the brine-bath is 23°, the quantity of heat passing per hour through 1 sq. ft. of the smooth walls from 0·06–0·08 of an inch thick may be taken as about 74,000 B.T.U., which corresponds with a coefficient of heat conduction $k=277$ B.T.U. per 1° difference of temperature.

For instance, with a cooling-machine with a yield of about 200,000 B.T.U. per hour, we may expect a condensation of 1·7 cub. ins. of water per second in the coil, which, for a velocity of 5·5 inches per second, gives a cross-section of 0·31 sq. in., or a diameter of 0·63 inch. With walls 0·06 inch in thickness, the mean heating surface of the tube (that is, the mean between the internal and external surfaces) is $2·074 \times 12 = 25$ sq. ins. (or 0·174 sq. ft.) per foot run, so that the length of coil in the brine must be

$$\frac{200{,}000}{74{,}000 \times 0·174} = \text{about 16 ft.}$$

It is of great importance in this method that the whole of the coil, as far as the vertical tube up which the condensed water passes, should have a sufficient slope and should afford no opportunity for the formation of water-pockets, which would undoubtedly freeze. The exit-tube should carry the water as quickly as possible out of the region of the cooling-brine, but it is convenient to make it quite vertical. At the beginning of the test a somewhat larger quantity of steam is allowed to pass than corresponds with the stable condition, and only when there is no danger of water collecting is the amount, and hence the pressure, regulated, so that the brine temperature remains at the required value and the water flowing out shows no signs of steam. This temperature is read off as often as possible—best every 5 minutes—on several thermometers and, as in the cooling test, the mean values are plotted graphically. A curve, *VV* (see Fig. 203), is thus obtained, and, in order to avoid troublesome and unreliable corrections, the end-point (at the close of the test) must be the same as the initial value. This can be easily arranged by regulating the amount of steam. The mean height of this curve then represents the mean temperature of the brine during the test.

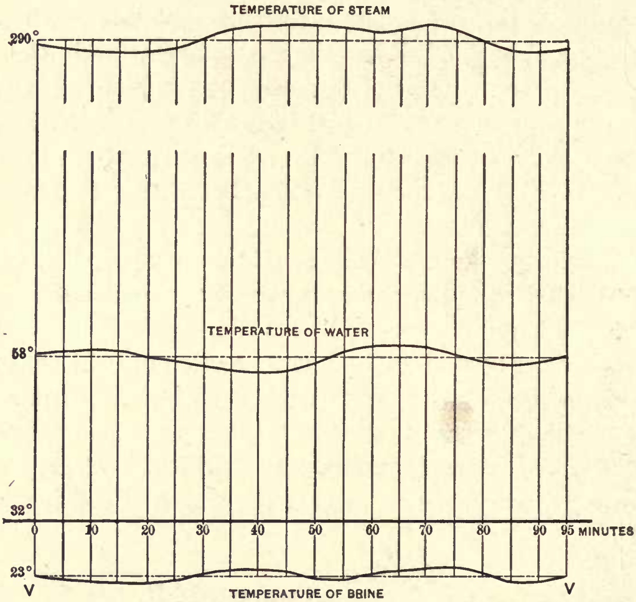

FIG. 203.

On the same diagram may be plotted the temperatures of the steam (obtained from the manometer readings) and those of the outflowing water. These temperature curves allow us to judge of the stability of the conditions of working, but it is also advisable to make frequent weighings, say every half hour, of the condensed water in the measuring-vessel. As the time of the test (which is generally limited by the capacity of the vessel on the balance) is never very protracted, it is unnecessary to represent these values graphically.

From the mean temperatures given by Fig. 203 or taken, with sufficient accuracy, as the arithmetic means of the observed values, it is now easy to calculate the cooling effect. If λ is the total heat of the steam corresponding with the condensation temperature t_1, t_0 the mean temperature of the outflowing water, and G the amount of the latter per hour, the net cooling effect is given by

$$Q=G(\lambda-t_0), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

which needs no correction of any sort. If, in addition, the gross cold generated is required, this may be obtained simply by a subsequent heating test, supposing that sufficient information is not given by the difference between the condenser yield and the equivalent of the compressor work.

If, for example, the temperature corresponding with the mean manometer reading of 42·7 lbs. per square inch excess pressure is $t_1=289°$ F., the total heat of the steam, according to Zeuner's Tables, is $\lambda=1170$ B.T.U. With an hourly condensation of $G=203$ lbs. of water, which at its exit from the spiral has the mean temperature $t_0=58°$ F., the net cold generated per hour is

$$Q=203(1170-58)=225{,}700 \text{ B.T.U.}$$

This process, which, indeed, necessitates a considerable amount of practice, is a very convenient one and gives very reliable results in a short time, and I am convinced that it has a great future.

44. The Calorimetric Investigation under Stable Conditions, which experience has shown to yield the most reliable results, presupposes that all the observed temperatures undergo no change during the whole duration of the test; that is, that the heat imparted to the cooling-machine in any interval of time, together with the equivalent of the compressor work done in the same time, agrees exactly with the heat given up. Since, however, in virtue of the to and fro motion of the piston, the compressor does not work uniformly but periodically, the smallest interval of time which can be taken must be the time of one rotation. The above condition means, therefore, that all the observed temperatures must have the same values at the end of a rotation as at the beginning. In consequence of the equalising action of the large masses occurring in the machine, especially that of the body (brine or cooling-water) which absorbs the heat, the fluctuations of temperature during one rotation of the compressor are always so small that they can only be detected with extremely sensitive thermometers and may hence be left out of consideration. In any case, they are always much smaller than those produced by irregular gains or losses of heat, which can never be entirely avoided in practical tests.

For the method in question this is of importance, since it becomes unnecessary to trouble about whether the compressor piston has exactly the same position or the same motion at the end of the test as at the beginning, although this would appear to be theoretically necessary.

After the initial and final times for the test have been chosen —or, better, settled from the course which things take during the test—the one matter requiring attention is the maintenance as far as possible of stable conditions. The process itself requires the exact measurement of the heat-absorbing liquids passing through the whole cooling-plant in a unit of time (hour) and the observation of the temperatures before and after the absorption of the heat. The taking of indicator diagrams of the compressor may serve for a control or for balancing the various amounts of heat. The most important determination is naturally that of the cold generated, which here—apart from air-cooling—is always to be measured in the brine. To this end the latter must pass through a reliable measuring-apparatus, best one with *Poncelet apertures*, i.e., a *vessel* furnished with sharp-edged circular holes, the height of the liquid in the vessel being determined as often as possible during the test.

The measuring-vessel is most conveniently introduced between the so-called brine return-pipe (i.e., the pipe through which the warmed brine is led back from the cellars or cold-store) and the evaporator, so that it stands above the latter. The brine then flows from the Poncelet apertures directly into the evaporator. In order to maintain a uniform head of liquid, unaffected by the influence of the incoming and outgoing brine, it is advisable to arrange in the vessel two cross-walls (see Fig. 204) which admit of communication between the three compartments so formed only at their lower extremities. The head is measured in the middle compartment, this being most simply done by means of an ordinary scale and by making use of an upper mark the height of which above the centre of the aperture has been previously determined.

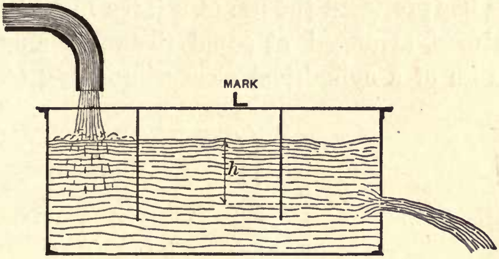

FIG. 204.

The means of measuring the head introduced in the Munich Experimental Station[1]—where the above-described method was used for the measurement of the condensed water of the steam-engine—by a point brought up to the level of the liquid from below, is of course more accurate than employing an ordinary scale, since it does away with the error due to capillarity; but apart from the time taken up in the exact adjustment of the point, which requires the whole attention of one observer, such an arrangement is not always at one's disposal in practice.

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[1] *Schröter*, Untersuchungen an Kältemaschinen verschiedener Systeme, II Bericht, 1890.

Further, it is often necessary to have the measuring-vessel made on the spot of soldered tin-plate, in which case holes cut in the walls have to replace neatly worked and accurately calibrated Poncelet apertures. Still, quite sufficient accuracy can be obtained in this way, especially if the precaution is taken of directly measuring the outflow for various heads with the help of a chronometer (which indicates to one-fifth of a second and can be stopped).

To facilitate the control of the amount of brine, Professor E. *Brauer*[1] has proposed the use of a large number of relatively small apertures, arranged at equal distances apart in a circle in the bottom of a cylindrical measuring-vessel (see Fig. 205), so that when the liquid enters axially the openings are symmetrically placed with respect to the incoming stream. By means of a funnel *T*, which is conveniently rotatable about a spindle, the liquid issuing from one of the apertures can be collected and led through a rubber tube into a vessel which has been previously calibrated, or can be subsequently weighed. The observation of the time during which the liquid is collected in this way by the funnel is the more accurate the smaller the amount

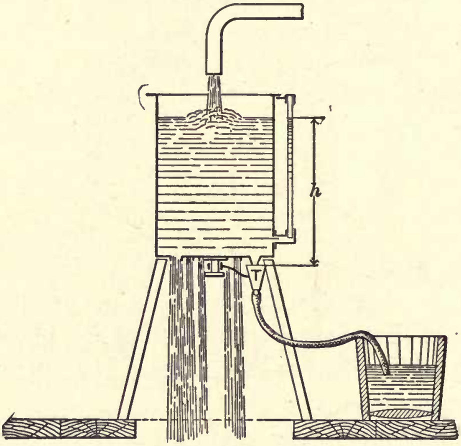

Fig. 205.

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[1] Zeitschrift d. Ver. D. Ingenieure, 1892.

of brine flowing through the opening. The head, h, is read off by means of a lateral gauge-glass as often as possible, but always at equal intervals of time. If this measurement is made for each aperture at least once during the test, the quantity of brine flowing out through all the holes in a unit of time is known, and so also is the so-called coefficient ot outflow.

The above method may, of course, be also employed for determining the amount of water passing through the condenser, but then the vessel is most suitably connected with the exit-pipe of the condenser, since there is always a sufficient fall in this case and none of the pressure mains need be unmounted for the test. Since, moreover, the determination of the condenser efficiency is easier when the inflow of water is regular, and, in conjunction with the estimated compressor work, permits of the cold generated being arrived at, we frequently find such Poncelet apertures connected directly with the condenser overflow (Fig. 206)—a practice introduced by the Linde Ice-Machine Company. These can then be only roughly designed, but they can be used with advantage in the measurement of the condenser yield in practical tests and so save part of the preliminary preparation.

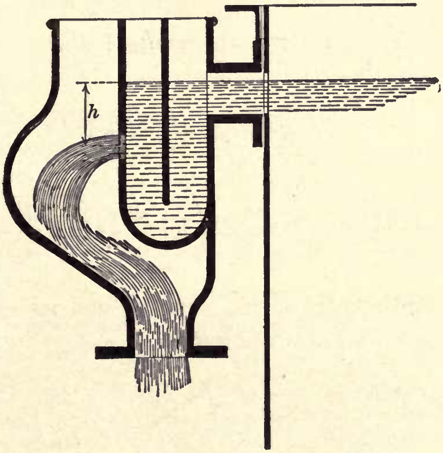

FIG. 206.

In the application of the data obtained, it is generally sufficient to use the arithmetic mean, h_m, of the observed heads in calculating the amount of liquid circulating, and the mean difference, t_m, between the temperatures on entering and leaving the evaporator, for calculating the cold generated. It will first be convenient to consider the errors occurring here. For this purpose, let us take c as the specific heat per gallon of the liquid and ϕ the coefficient of outflow[1] of the apertures, both of these values being taken as constant over the intervals met with in practice. If n is the number of separate observations and F the total cross-section of all the apertures, the cold generated is given by

$$Q=\frac{\phi.c.F}{n}.\Sigma(t.\sqrt{2gh}), \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where t is one of the observed differences of temperature and h the corresponding head. If, now, we take

$$t=t_m+\Delta t \quad \text{and} \quad h=h_m+\Delta h,$$

we have

$$Q=\frac{\phi c.F}{n}.\sqrt{2g}\Sigma(t_m+\Delta t)\sqrt{h_m+\Delta h}$$

$$=\frac{\phi cF}{n}.t_m\sqrt{2gh_m}.\Sigma\left(1+\frac{\Delta t}{t_m}\right)\sqrt{1+\frac{\Delta h}{h_m}}$$

If, now, the deviations from the mean values are small, we may take this expression as approximately

$$Q=\frac{\phi cF}{n}.t_m\sqrt{2gh_m}\Sigma\left(1+\frac{\Delta t}{t_m}+\tfrac{1}{2}.\frac{\Delta h}{h_m}\right), \quad . \quad . \quad . \quad (6)$$

where, on the one hand, the product of $\frac{\Delta t}{t_m}$ and $\frac{\Delta h}{2h_m}$ and, on the other, all higher powers of $\frac{\Delta h}{h_m}$ are neglected. But since t_m and h_m represent arithmetic means,

---

[1] For this I have obtained, for sharply cut Poncelet apertures, the values 0·620–0·635, which are in agreement with other results. The specific heat of the brine is determined from Table XVII, on page 141, after ascertaining the specific gravity with an aërometer

$$\Sigma\Delta t = o \quad \text{and} \quad \Sigma\Delta h = o,$$

so that, for n observations, the expression in the brackets reduces to

$$\Sigma(1) = n,$$

and, instead of (6), we have

$$Q = \phi . c . F . t_m \sqrt{2gh_m}; \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

i.e., *with small fluctuations of the observed data from their mean values, or, what is the same thing, very constant conditions during the test, the calculation of the cold generated may be based on the mean values.*

If any one is not quite clear as to the reliability of this simplified method, the only thing to be done is to calculate the separate values of the product given in brackets in equation (5) for each single difference of temperature and the corresponding head, and then to add all these together. That it is useful to record all the observed data graphically will be readily understood from what has gone before. The constancy of the conditions is especially easily judged of in this way.

In order to show the accuracy of the method of calculation just described, we shall test it on a series of observations. The following table (page 262) contains the observed heads, h, their square roots, $\sqrt{h}$, the corresponding differences of temperature, t, and, lastly, the products $t\sqrt{h}$.

If, now, we calculate the value of the product, $t_m\sqrt{h_m}$, from the mean values of the first and third columns, we get

$$\sqrt{h_m} = \sqrt{8\cdot93} = 2\cdot988 \quad \text{and} \quad t_m\sqrt{h_m} = 36\cdot0 . 2\cdot988 = 107\cdot6,$$

while the exact value, as given in the last column, is 107·4. So

	h	$\sqrt{h}$	t	$t\sqrt{h}$
	9·80 inches	3·130	35·6° F.	111·4
	8·70 "	2·950	36·1	106·5
	7·99 "	2·827	36·3	102·6
	8·54 "	2·922	36·1	105·5
	9·45 "	3·074	35·8	110·0
	9·13 "	3·022	35·8	108·2
Mean values. . .	8·93 "	2·988	36·0	107·4

that, in this case, where the separate readings of both the head h and the temperature-difference t deviate by 10 per cent from the mean values, the error is only about 0·2 per cent and can therefore be neglected, since in all such practical tests the direct errors of observation have much higher values. The method of calculating from the mean values is hence applicable within fairly wide limits. In view of the widespread use of this method, the foregoing proof may perhaps appear to be superfluous; but it is always useful if the investigator can, in every separate case, demonstrate scientifically the reliability of the methods of observation and the application of the observed data in determining the final result. This precaution is, as I know from experience, not always observed and then has sometimes very doubtful consequences in practice. If very great fluctuations of the head (as much as half the mean value) occur during the test—as is not infrequently the case with pumps giving an irregular supply—generally accompanied by considerable variation in the temperature-difference, the calculation should not be made with the mean values of the separate readings, but the product $t\sqrt{h}$ should be determined for each observation and the cold generated obtained from these values. The graphic representation of the readings, made during the test if possible, simplifies the supervision to an extraordinary degree.

The employment of a measuring-vessel with outflow apertures depends, as we have seen, on the possibility of determining the head from time to time during the test. With *frothing salt solutions*, with which I have met now and again, this cannot be done with certainty, so that, unless the measuring-vessel is provided with a lateral gauge-glass in which the surface of the liquid is free from froth, the whole method becomes impracticable. In such cases we may often use a *float* which protrudes above the froth, and the depth of immersion of which has been previously determined in another vessel filled with the same brine (which, when at rest, exhibits no foam) or in the measuring-vessel itself with the apertures closed. Unfortunately, the float is often kept moving to such an extent by the vigorous evolution of bubbles that it is difficult, if not impossible, to obtain satisfactory readings of the head; it is then best to use a quite different method.

The most exact method and the one most free from objection consists, of course, in weighing the brine by means of two vessels standing on balances. Even with medium-sized cooling-plants these vessels must have such a large capacity that it seems almost impossible to construct them and the balances for weighing them. On this account I have always decided to collect the quantity of liquid returning in a certain period of time, say every 20 minutes, in a vessel with, as nearly as possible, a uniform cross-section and to determine the time taken in filling by means of a chronometer. In breweries, where such investigations have most frequently to be made, such vessels are almost always available in the form of cylindrical hop-boxes holding from 60 to 70 gallons. After fixing at the bottom of the vessel (Fig. 207) an ordinary cock for the purpose of emptying it after the measurement, the vessel is placed over the evaporator, so that the return-pipe for the brine is just above it. A three-way cock is then introduced into this pipe, so as to allow the brine to pass into either the evaporator or the vessel. The measurement itself requires an observer equipped with a chronometer and a workman to open and shut the three-way cock. This opening and shutting is done at the command of the observer, who at the same time starts and stops the chronometer. The temperature of the incoming brine can also be determined during the filling, while the temperature at exit is best taken immediately before the measurement. This is done on account of the slight disturbance of the stable conditions of the evaporator, from which liquid is continually being pumped during the filling. Since there is no inflow, the temperature of the outflow must fall slightly, thus making the cooling effect appear greater than it really is. By reading the temperature before measuring the liquid, this error is avoided.

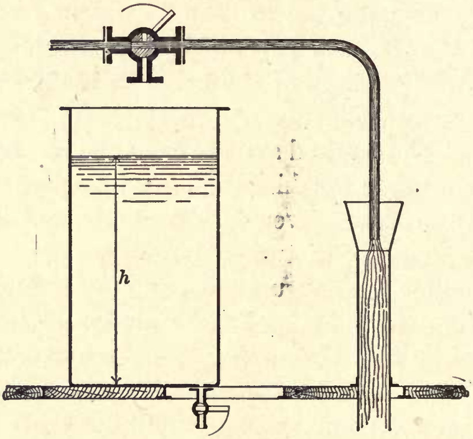

FIG. 207.

After the vessel is filled and the froth removed, the determination of the height of the liquid offers no difficulty. If we know from a preliminary or subsequent weighing of the vessel both empty and when filled with brine up to about the mean height, the separate values of the height, h, the cross-section, F, and the observed time, τ, give the volume per unit of time as

$$V=F\cdot\frac{h}{\tau}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

or, if we again introduce for the mean values, h_m and τ_m, the expressions

$$h=h_m+\Delta h \quad \text{and} \quad \tau=\tau_m+\Delta\tau,$$

we have approximately, where the fluctuations Δh and $\Delta\tau$ are small,

$$V=F\cdot\frac{h_m}{\tau_m}\cdot\frac{1+\frac{\Delta h}{h_m}}{1+\frac{\Delta\tau}{\tau_m}}=F\cdot\frac{h_m}{\tau_m}\left(1+\frac{\Delta h}{h_m}-\frac{\Delta\tau}{\tau_m}\right). \quad . \quad . \quad . \quad (8)$$

If we take h_m and τ_m as the arithmetic means, the values of $\Sigma\Delta h$ and $\Sigma\Delta\tau$ disappear on summation and we obtain

$$V=F\cdot\frac{h_m}{\tau_m}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8a)$$

i.e., *with small fluctuations of the height and time of filling of the measuring-vessel from their mean values, the quantity of circulating liquid can be determined solely from these mean values.* That the mean value of the temperature difference may also be taken follows from the fact that, as we have already seen, its measurement cannot take place at exactly the same time as that of the quantity of liquid. If, then, c represents the specific heat of the solution per cubic foot and t_m the mean difference of temperature in degrees Fahrenheit, and if F is in square feet, h_m in feet, and τ_m in seconds, the cold generated per hour is

$$Q=3600.F\cdot\frac{h_m . t_m}{\tau_m}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

there being no necessity to introduce coefficients. To judge of the accuracy of the above-described calculation of the quantity of circulating brine from the mean values observed, we may employ the readings given in the following table. The first column gives the heights of liquid in the measuring-vessel in inches, the second the times of filling in seconds, and the third the quotient of these two; the values in the last column indicate a very uniform supply by the pumps.

The quotient $h_m : \tau_m$ has the value 1·205, which differs from the exact number, 1·204, by less than 0·1 per cent, a difference which may be neglected. Further, when the supply of brine by the pumps is moderately regular, the observer can keep

	h	τ	$h:\tau$
	29·1 inches	24·2 seconds	1·202
	28·4 "	23·6 "	1·203
	27·9 "	23·2 "	1·203
	29·6 "	24·4 "	1·213
	29·3 "	24·4 "	1·201
	28·6 "	23·8 "	1·202
Mean values.......	28·8 "	23·9 "	1·204

the fluctuations of the separate readings from the mean values very small by giving the word for the cock to be closed after the lapse of a definite time. Quite satisfactory results can be obtained by this method, devised to meet an emergency.

A similar method is also frequently advisable for determining the condenser effect in cases where, for some reason or another, the use of Poncelet apertures is inadmissible. The cooling-water is passed, before it enters the condenser, into a reservoir, in which the fall of level in a certain time is measured after the supply is cut off. If no reservoir is obtainable, the cooling water may be led first of all into an elevated cistern, e.g., a fresh-water cooler, in which the measurement may be made after the supply has been stopped or led into another cistern. With two such elevated cisterns, which should be as similar as possible, the whole of the cooling-water may be measured. Besides an exact knowledge of the cross-section of the cisterns, this method requires a careful regulation of the supply to the condenser owing to the variable height of the water in the cisterns.

If it is found by a preliminary experiment that, *owing to the excessive action of the machine in relation to the sources of heat, constant conditions cannot be attained* or permanently maintained, an attempt must be made to compensate for the excessive cold generated by the introduction of a fresh source of heat. There are several means of doing this, the simplest being the arrangement of a heating-coil, fed with warm water or steam, in the measuring-vessel shown in Fig. 205. If, then, stress is laid on the determination of the excess of cold generated when the working is regular, it can be obtained from the quantity and temperature-difference of the inflowing and outflowing water or steam, in the latter case the latent heat being taken into account. Where no suitable heating-coil—that is, one that is sufficiently rigid and tight—is accessible, we may, at least when the amount of brine in the pipes and evaporator is large, pass *direct steam* into the measuring-vessel. Since more than 1000 B.T.U. are set free by the condensation of 1 lb. of steam, such a very small quantity of the latter is necessary that the specific heat of the total circulating brine is not appreciably altered. This can be shown by determining the specific gravity of the salt solution before and after the test, since this magnitude is very intimately related to the specific heat. The passage of the steam into the brine must naturally be exactly regulated by a throttle-valve and temperature observations, and, if it enters the middle chamber of the measuring-vessel shown in Fig. 205, we can determine the amount of heat introduced by noting the temperatures of the brine on entering and leaving the vessel. The steam used is only approximately given, because its water-content, when it is introduced into the brine, cannot be regarded as known. It is here not worth while, as is done in *Pressl's* process, to free the steam from water.

AMERICAN PRACTICE

IN

REFRIGERATION

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IN

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BY

H. M. HAVEN, S.B.
Member of the American Society of Mechanical Engineers and of the American Society of Refrigerating Engineers

AND

F. W. DEAN, S.B.
Member of the American Society of Mechanical Engineers

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PREFACE.

THE determination of the Publishers to present a translation of Lorenz's Neuere Külmaschinen to the American public has furnished a good opportunity for them to present also several brief chapters on some of the salient features of refrigerating practice in the United States. It has been the purpose of the authors to deal with this as briefly as possible and yet furnish the principal information which a reader would be likely to seek, and thereby to place him upon the right track in any work which he may undertake. There are, of course, many features and methods of refrigeration which the scope of our work does not cover, and these the novice will devise to the best of his ability and the experienced man will do almost instinctively. It has become the custom of technical papers to illustrate and describe manufacturing plants of all kinds. and readers of the papers devoted to refrigeration will find therein many instructive accounts of all features of such work. In fact, no practitioner can in these days be well equipped in his department of engineering without reading to some extent the periodical literature of the day. These publications must to a great extent replace books, but of course books are convenient places to collect and store information and thereby save time and space for the student and engineer. It is principally for this reason that scientific books are purchased, and we shall be glad if our modest attempt receives a share of space in the libraries of refrigerating engineers.

THE AUTHORS.

BOSTON, November 1904

INTRODUCTION.

In contradistinction to the German practice, we shall illustrate in a concise way the methods and machinery used for Refrigeration and Ice-making in the United States.

While carbonic acid, sulphurous acid, and cold-air machines are sometimes used on shipboard, in hotels, and in small units elsewhere, ammonia refrigerating machines, both compression and absorption types, are the principal agents in installations of any size throughout the United States.

AMERICAN PRACTICE IN REFRIGERATION.

CHAPTER I.

GENERAL METHODS OF REFRIGERATION.

REFRIGERATED warehouses and compartments are being established in nearly all of the large business centres throughout the United States, for the preservation of fruits, meats, dairy and other products. These compartments, in con-

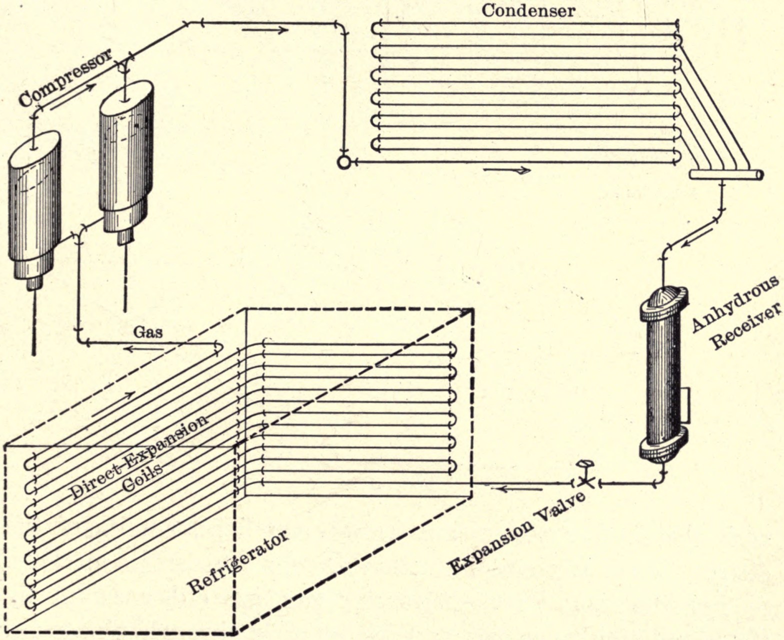


FIG. 208.

junction with refrigerator-car lines, admit of the economical distribution of perishable products to any centre, drawing from the section of greatest supply and least demand and delivering to the section of restricted supply and greater demand. This, with the further connecting trans-ocean lines, admits of the distribution of any overproduction or supply to foreign markets.

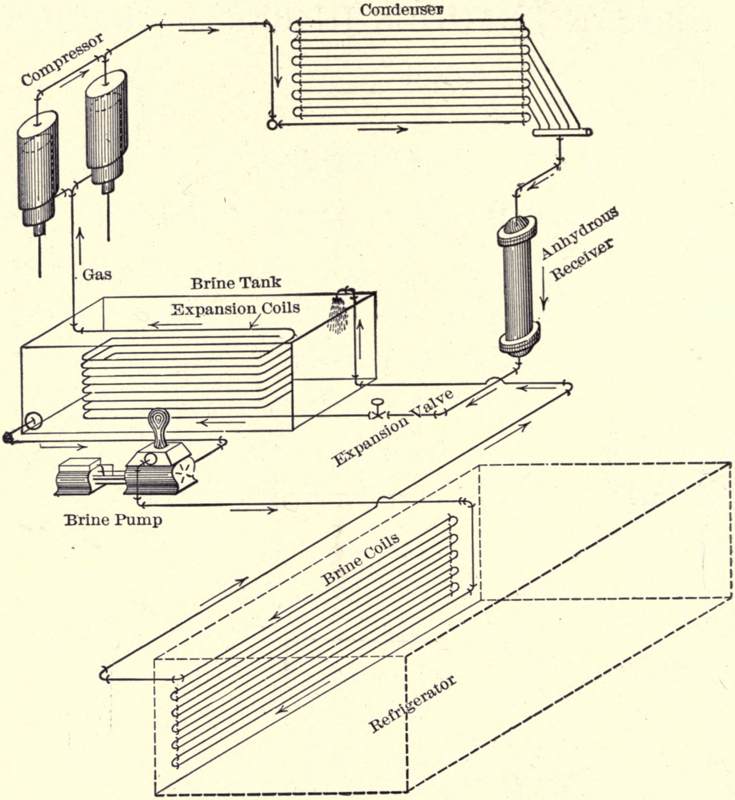


FIG. 209.

For the most part refrigerator-cars are either cooled by the circulation of a salt and ice mixture through pipe coils, or directly by ice in bunkers, while in marine practice refrigerating machinery is installed as described under marine refrigeration.

In refrigerating warehouses the several methods of refrigeration commonly used may be described as "Direct Expansion," "Brine Circulation with Brine-tank," "Brine Circulation with Brine-cooler," "Indirect Air Circulation with

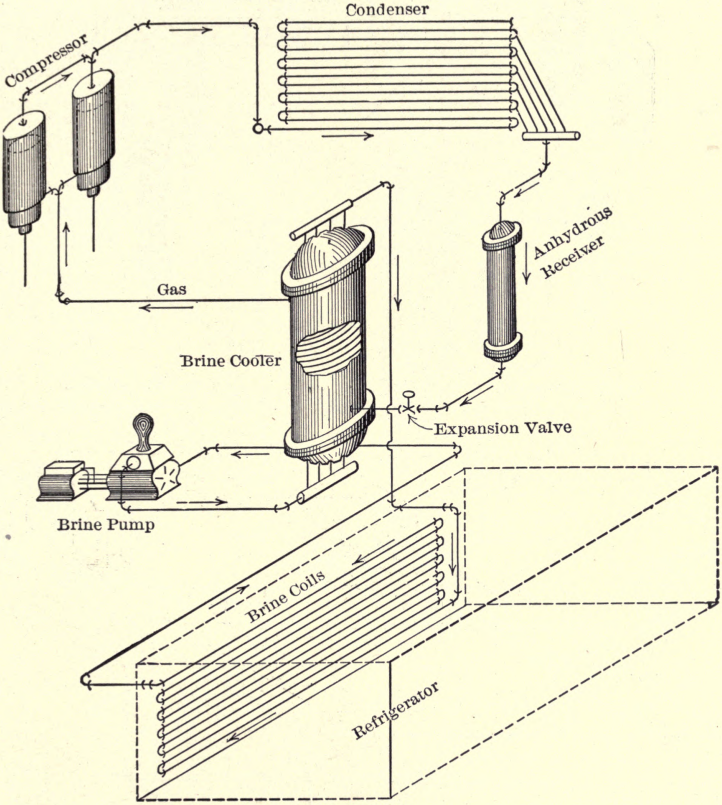


FIG. 210.

Direct-expansion Bunker Coils," and "Indirect Air Circulation with Brine Bunker Coils."

1. **Direct Expansion,** as in Fig. 208, is among the first methods adopted, and is still used extensively in fish, meat, general freezing, and in ice-making.

The liquid ammonia from the receiver is expanded through one or more expansion-valves to the coils in the refrigerators, the heat from the materials in these compartments vaporizes the liquid ammonia, and as a gas it passes to the refrigerating-machine, which, if it is a compression machine, compresses it and passes it to the condenser, where it is condensed. It is finally collected in the receiver ready for re-expansion.

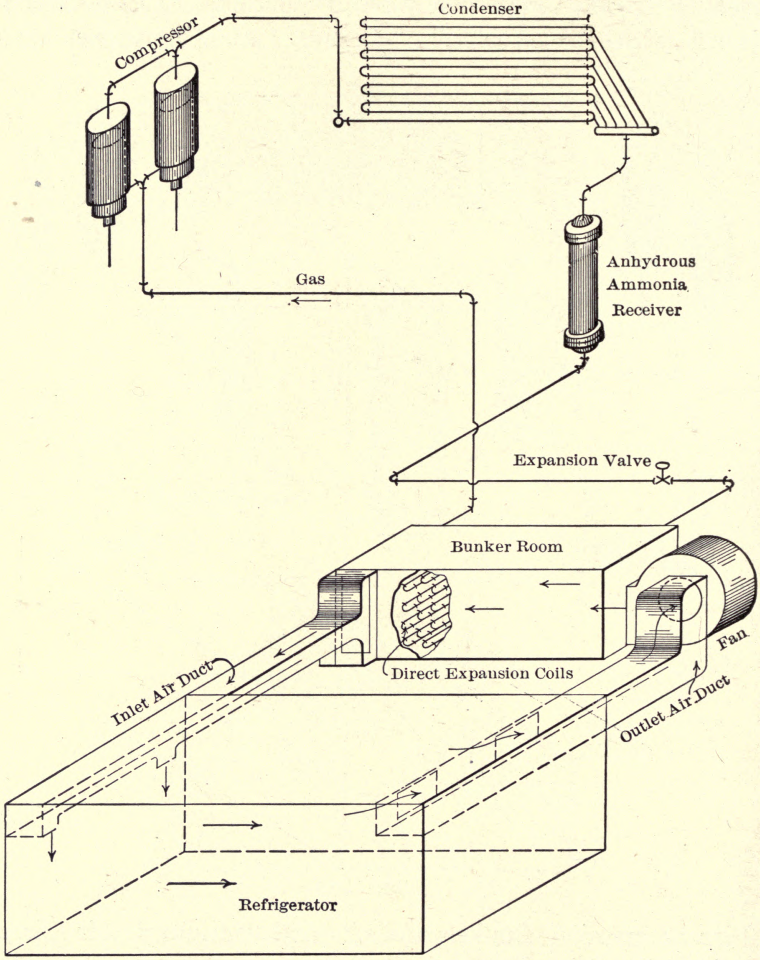


Fig. 211.

2. **Brine Circulation with Brine-tank,** as in Fig. 209, has the direct-expansion coils immersed in a bath of brine which

is circulated through the tank to the brine-coils in the refrigerator, and back again to the tank. All ammonia-coils are thus eliminated from the refrigerator itself.

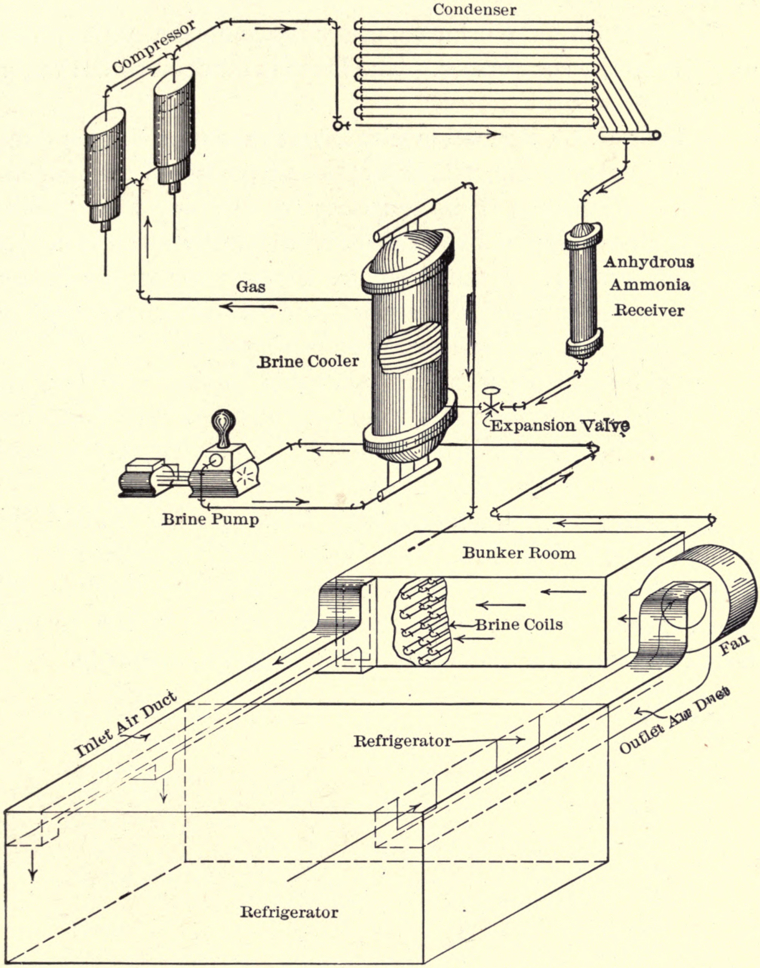


FIG. 212.

3. **Brine-circulation with Brine-cooler,** as in Fig. 210, is a step further in simplifying the manipulation of the ammonia by replacing the many expansion-valves of the direct-expansion

coils by a fewer number, often by one such valve which expands the ammonia from the receiver into a shell, and around a series of spiral pipe coils containing a calcium-brine solution in circulation.

The heat from the brine vaporizes the ammonia, and it passes as a gas to the compressor. The brine is pumped to the refrigerator coils as before.

4. **Indirect Air-circulation with Direct-expansion Bunker Coils.**—This is shown in Fig. 211, and takes advantage of the increase in heat-transmission from surfaces in contact with rapidly moving air-currents above that from equivalent surfaces in contact with still air. A fan circulates the air through the bunker coils to the refrigerator, and again to the coil-room for re-cooling.

All piping is removed from the refrigerator.

Should leaks occur in the ammonia piping, however, the rapidly moving air-currents would quickly carry the ammonia-gas to the refrigerator itself. In order to prevent such a possibility the following method was devised.

5. **Indirect Air-circulation with Brine Bunker Coils.**—This is illustrated in Fig. 212. It is being rapidly adopted for fruit and egg storage, and promises to be extensively used for the preservation of all perishable products. It is the method usually adopted for cooling auditoriums and apartments.

CHAPTER II.

REFRIGERATED COMPARTMENTS.

ALL of the methods just described are being more or less successfully used in refrigerated warehouses and in compartments for cold storage, which, according to the temperatures maintained, are classified as "Coolers," having a temperature of 30° F. or higher, as "Holding Freezers," having a temperature of 10° F. or lower, and as "Sharp Freezers," having a temperature of 0° to 20° below zero.

Perishable goods are kept in each of these divisions, and the temperatures maintained by the warehousemen in the Eastern States are as follows:

1. Cooler:

Kind of Goods.	Temperature.
Provisions, fish, and meat	34°–36° F.
Vegetables	34 –36
Onions	31 –32
Dried fruits	35 –36
Buckwheat flour	35 –36
Oranges	40 –41
Apples, summer	33
" winter	31 –32
Ferns	32 –33
Eggs	31 –32
Cheese	31 –33
Nuts	36 –38
Pears	35 –38
Beer, summer	34 –35
" winter	38 –40
Meats, poultry, and game	29 –30

2. Holding Freezers:

Kind of Goods.	Temperature.
Butter	0° F.
Poultry	10
Fish	10
Meat	10
Frozen eggs	10

3. Sharp Freezers:

Kind of Goods.	Temperature.
Butter	0°– 5° F. below zero or lower
Poultry	0 – 5 " " " "
Fish	5 –10 " " " "
Eggs in cans	10 " " " "

4. Warehouse Capacities.—There has been a tendency of late to increase the size of refrigerated compartments until, in some large warehouses which have been built within the last few years, the units for egg and fruit storage have a capacity of from 65,000 to 70,000 cu. ft., while units of 50,000 cu. ft. or thereabout are quite generally adopted. There has been some doubt expressed as to the possibility of maintaining uniform temperatures and humidities throughout such large rooms. This is being successfully done, however, where the indirect air-circulation is used, if proper insulation and air-locks are installed. Air-locks are adopted at the entrance to each room, one door being opened and closed before the other is used.

It has been found impossible to maintain the proper humidity in egg-storage rooms without this air-lock, more moisture otherwise leaking into the room at each opening of the door than can be removed by the cooling-coils and drying-racks.

In sharp freezing the tendency has been to adopt rooms of from 10,000 to 15,000 cu. ft. or smaller.

There is a growing tendency with the adoption of better insulation to use freezers of 40,000 cu. ft. or larger; some of 65,000 cu. ft. capacity recently have been used where a combination of direct piping with the indirect circulation of air is installed. The size of a room depends, however, upon the character and quantity of the goods stored.

In holding freezers the tendency is to adopt large units.

In pan-fish freezing a number of small units very heavily piped, as described later, is used in preference to large units.

5. **Piping Required.**—In piping compartments, while exact conditions need to be known for each case, general rules can be given which may be safely used for approximate determinations.

For brine-piping, on a basis of utility and efficiency, coils of 2-in. common wrought-iron pipes made up with return-bends 6 in. on centres or thereabout are desirable.

For indirect circulation with brine bunker coils and good insulation, a ratio of 1 lineal foot of 2-in. pipe to 30 cu. ft. of storage space will maintain fruit and egg compartments at 31° F. A ratio of 1 to 40 is sometimes used.

For direct brine-piping and good insulation,

the ratio for 0° F. temperature is 1 lineal foot of 2″ pipe to 3 cu. ft. space
" " " 10° " " " 1 " " " " " 6 " " "
" " " 20° " " " 1 " " " " " 8 " " "
" " " 30° " " " 1 " " " " " 10 " " "
" " " 35° " " " 1 " " " " " 12 " " "

For direct-expansion piping,

the ratio for 0° F. temperature is 1 lineal foot of 2″ pipe to 5 cu. ft. space
" " " 10° " " " 1 " " " " " 10 " " "
" " " 20° " " " 1 " " " " " 15 " " "
" " " 30° " " " 1 " " " " " 20 " " "
" " " 35° " " " 1 " " " " " 25 " " "

For ice-storage with brine-piping the usual ratio is 1:20.

For auditoriums and similar cooling work the requirements are so varied and exacting that only a careful determination for each case will give satisfactory results.

All of the above ratios are for large rooms; for smaller compartments, such as are found in hotels, restaurants, market-boxes, and similar places in exposed locations, the following ratios may be used successfully:

For capacities of	100– 500	cu. ft. a ratio of	1:2
" " "	500– 1,000	" " " " "	1:4
" " "	1,000– 5,000	" " " " "	1:5
" " "	5,000–10,000	" " " " "	1:6
" " "	10,000–15,000	" " " " "	1:7

These ratios with a brine temperature of 15° will maintain the compartments at 40° or lower. The insulating should be at least as good as $\frac{7}{8}$-in. double sheathing with paper between, 4 in. of mill shaving or, better, granulated cork, and $\frac{7}{8}$-in. double sheathing with paper between.

6. Refrigeration Required. — The refrigeration required depends upon the temperature to be maintained in the compartment, its character, whether a cooler or freezer, the exposure, and whether the refrigeration is constant or intermittent. The exposure depends upon the ratio between the cubical contents and the exposed surfaces of the refrigerator, the insulation, and the frequency with which the air is renewed.

It is usual, in estimating the work which one ton of refrigeration will do, to assume the following as fair approximations, each case being computed separately for exact results:

In refrigerated warehouses one ton of refrigeration in twenty-four hours will maintain

10,000	cu. ft. of general storage space at	35°–40° F	temperature
6,000	" " " egg " " "	30°–32° "	"
3,000	" " " butter " " "	15°–20° "	"
2,000	" " " game and poultry " "	10° "	"
1,500	" " " " " " " "	0° "	"

This is for average conditions with good insulation and for plants requiring 50 tons of refrigeration or more.

In breweries one ton of refrigeration will maintain 8000 cu. ft. of general storage space at 30° to 36° F., or will cool 40 barrels of beer-wort from 70° to 40° F.

In abattoirs and packing-houses one ton of refrigeration will maintain

10 000	cu. ft of curing space at	35°–40° F.
3.000	" " " freezing " "	20° "
1,500	" " " " " "	0° "

If the number of animals is known, independent of the heat-losses through walls and exposed surfaces, one ton of refrigeration is required to properly chill

7–10 beeves, each weighing about 700 lbs., and surrounding space
20–25 hogs “ “ “ 250 “ “ “ “
50–60 calves “ “ “ 90 “ “ “ “
70–75 sheep “ “ “ 75 “ “ “ “

If the exposure is excessive, more refrigeration will be required. In small units, such as compartments in hotels, markets, and similar places where the exposures are large relative to the size of the room, and the door-openings are frequent, it will often require one ton of refrigeration to maintain 250–300 cu. ft. of storage space at 35° F. or thereabout.

It is assumed in all of these quantities that the insulation is sufficient to prevent excessive heat-leakage.

7. Government Investigations.—An investigation of two or more years' standing is in progress under the direction of the United States Department of Agriculture, to secure data which shall determine among other things the proper temperatures and humidities to be maintained in warehouses for perishable products.

This department has made a wide investigation of the preservation of apples, pears, peaches, and other small fruits, not only from the standpoint of warehousemen, but of the orchardist and fruit-handler. These investigations are important as influencing the design of the warehouse and fixing proper conditions for refrigerated compartments intended for the care of perishable goods.

It has been found from the orchardist's standpoint that the environment of the orchard, such as soil and general location, affect the color, size, form, quality, and texture of the fruit, also that bacteria or fungi which produce rot are affected by local conditions.

From the standpoint of the fruit-handler and warehouseman the ideal condition in fruit-storage is reached when the fruit is placed in the warehouse in the quickest time possible, after picking, in packages that cool down quickly.

The temperature should be low enough to prevent the ripening and spreading of such diseases as fungi, bacteria, or scald, and a temperature of 31° or 32° F. has been found desirable for this purpose.

The degree of cold in the warehouse is not the only factor in fruit-preservation. Deterioration in flavor and texture is often due to impure air and to improper regulation of humidity. Low temperatures are necessary for quick-ripening fruits.

Wrapping fruits in waxed or other papers prevents the spread of disease from one fruit to another. At the present time experimental investigation is being conducted relative to dairy products to decide the best temperature and humidity for each and the length of time each can be safely stored.

An investigation is also on foot to determine the effect upon the human system of food that has been in cold storage different periods of time.

Neither of the later investigations has progressed far enough for the publication of results.

CHAPTER III.

INSULATION.

1. **Characteristics.**—With the increased application of refrigeration, the importance of properly insulating refrigerated compartments against the entrance of heat through exposed wall-surfaces has become more apparent. It is evident that wherever material stored in a properly insulated compartment is once cooled, only enough additional refrigeration is required to compensate for losses due to the entrance of heat from without. In the last few years experiments have been made upon a great number of materials suitable for insulating purposes.

Among the best commercial insulators are the following: hair-felt, granulated cork, nonpareil cork, mineral wool, rock wool, lith, pumice, cork and pitch, and mill shavings. These substances, in different combinations with wood, brick, cement, and air-spaces, have been used for the insulation of refrigerated compartments and warehouses structures.

While it has been demonstrated that the dead-air space is the best non-conductor of heat, the size to which it must be reduced in order to prevent any circulation within it is in dispute. It is apparent that the smaller and more numerous the air-spaces are, the better.

If a large air-space exists between two vertical exposed surfaces of different temperatures, the air on the cold side descends, while that on the warm side rises and a circulation is thus created which insures a rapid transmission of heat from one surface to the other.

If, however, this space is divided vertically into several sections, the transmission is decreased, and if, further, these later divisions are again horizontally subdivided, still better results are obtained. It has been demonstrated that insulators of the best non-conductivity are obtained when such a division of the air-space is accomplished by filling it with some material such as granulated cork or mill shavings, the interspaces forming minute dead-air cells having practically no communication with each other.

The value of the insulator is impaired either when the particles composing it are so wide apart as to insure a communication between the interspaces, or when the material is so compressed or gummed together that either infinitesimal or no air-spaces at all exist between these particles. It has been fairly well shown that if heat in passing from one exposed surface to another is obliged to pass through strata of different densities, a loss of heat apparently takes place at each change of density. The foregoing properties hold true only when the insulation is kept dry by making it air-tight and moisture-proof.

An insulation, however good when new, will depreciate rapidly if it becomes damp by an inflow of moisture. Besides being moisture-proof, insulation must be non-odorous, unlikely to settle, to decompose, or to disintegrate. It should also be slow-burning or fire-proof, and easily applied. Recently insulating papers of different kinds have been adopted in insulated structures. Their value depends not upon their insulating value, but upon their efficiency in intercepting moisture and making the construction air-tight. There is a probable additional value in making laminations which change the density of the strata, thus retarding the flow of heat somewhat.

In order to prevent the penetration of moisture into the insulation, other materials, such as pitch, paraffin, and many kinds of so-called water-proof paints and varnishes, are used.

2. **Types.**—The types of insulation shown in Figs. 213–227 have been used by prominent engineers in large and important plants, in the light of experiments and experience. These insulations need to be compared relative to their first cost and insulating values.

The types Figs. 213–217 have been used for the exposed walls of refrigerators whose temperature is likely to be 0° F. or lower on one side and 80° F. or higher on the other.

The types Figs. 218–221 are of floor construction with the same differences of temperature.

In basement rooms where the inside temperature is 0° F. or lower, the types Figs. 222–224 have been adopted.

Constructions similar to those shown by Figs. 225–227 have been employed in partitions between corridors and rooms having a temperature of 0° F. or lower.

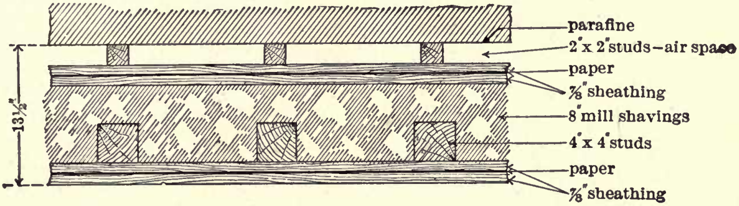


FIG. 213.

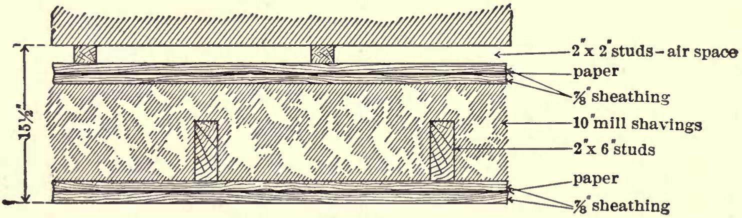


FIG. 214.

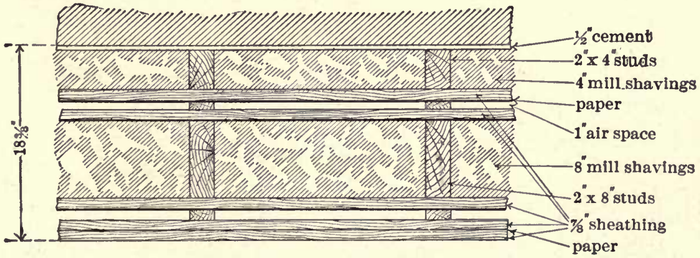


FIG. 215.

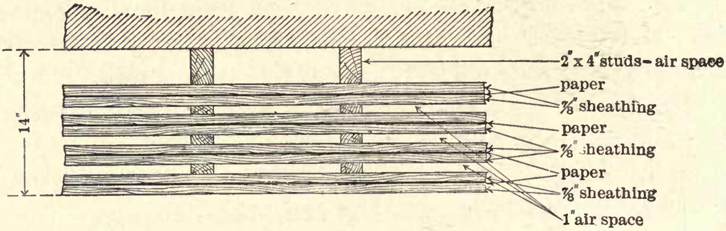


FIG. 216.

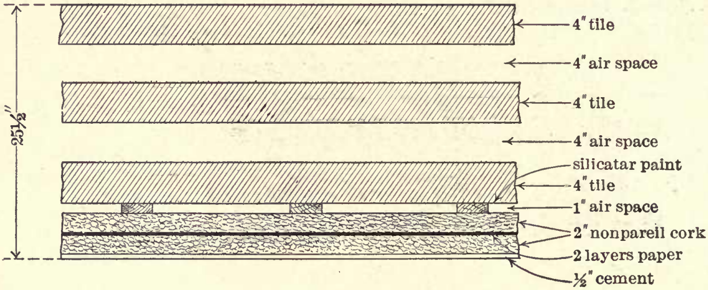


FIG. 217.

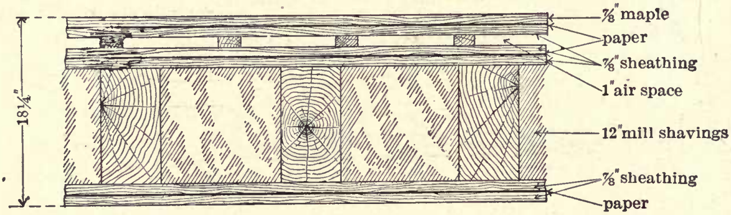


FIG. 218.

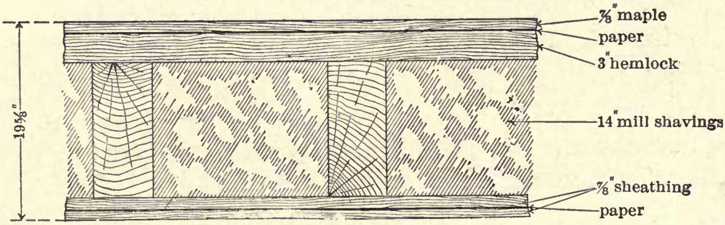


FIG. 219.

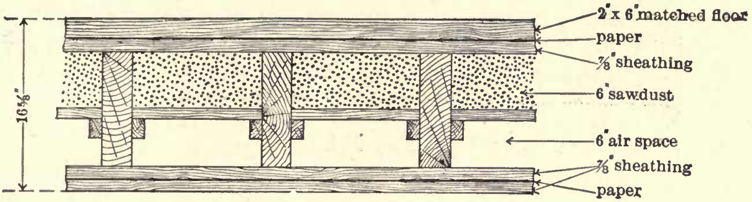


Fig. 220.

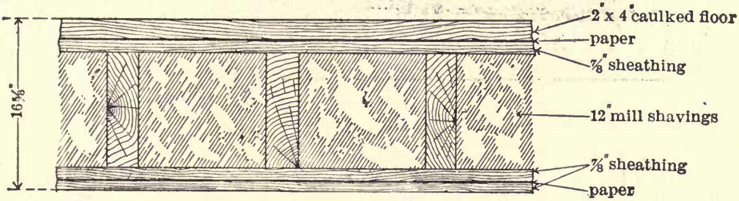


Fig. 221.

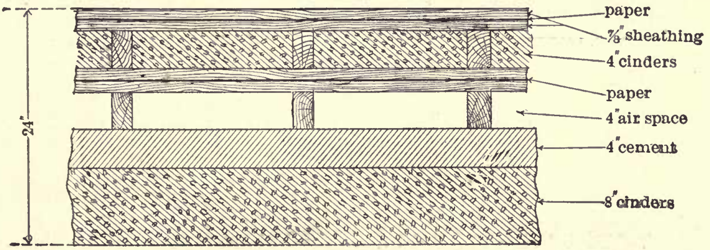


Fig. 222.

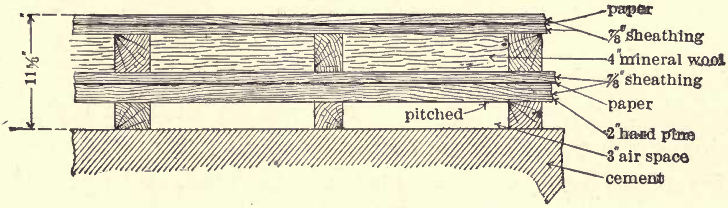


Fig. 223.

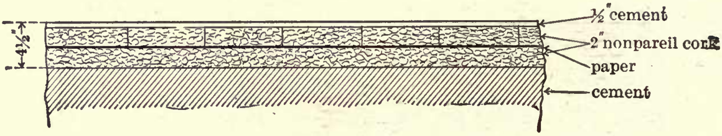


FIG. 224.

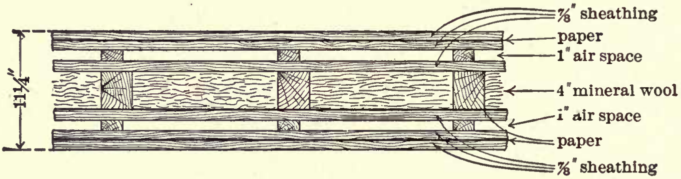


FIG. 225.

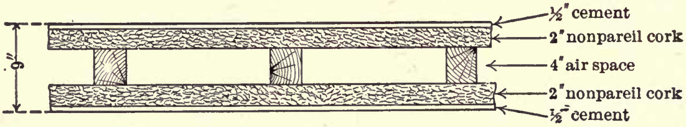


FIG. 226.

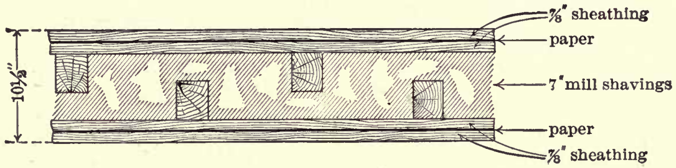


FIG. 227.

3. **Insulation Design.**—Peclét in his "Traité de La Chaleur" describes exhaustive experiments which he made on the transmission of heat. Constants which he obtained for the coefficients of conductivity C of various materials reduced to English units are as tabulated below.

TABLE GIVING THE AMOUNT OF HEAT, *C*, IN B. T. U. PER HOUR, WHICH WILL PASS ONE SQUARE FOOT OF VARIOUS MATERIALS, PER ONE INCH OF THICKNESS, PER ONE DEGREE FAHRENHEIT.

	Density.	*C.*
SOLID MATERIALS.		
Marble, gray, fine-grained	2·68	28·1
“ white, coarse-grained	2·77	22·4
Limestone, fine-grained	2·34	16·8
“ “	2·27	13·6
“ “	2·17	13·7
“ coarse-grained	2·24	10·6
“ “	2·22	10·2
Plaster, ordinary	2·22	2·67
“ “, very fine	1·25	4·20
Brick	1·98	5·56
“	1·85	4·11
Fir (wood), transmission perpendicular to fibres	·48	·75
“ “ “ parallel to fibres	·48	1·37
Walnut, transmission perpendicular to fibres	·48	·83
“ “ parallel to fibres	....	1·40
Oak, “ perpendicular to fibres	....	1·70
Cork	.22	1·15
India-rubber	·22	1·37
Gutta-percha	·22	1·39
Glass	2·44	6·05
“	2·55	7·10
MATERIALS IN A POWDERED STATE.		
Quartz sand	1·47	2·18
Brick-dust, large grains	1·00	1·12
“, passed through a sieve of silk	1·76	1·33
“, fine powder obtained by decantation	1·55	1·13
Chalk, in powder, slightly damp	·92	·870
“ “ “ washed and dried	·85	·694
“ “ “ washed, dried, and compressed	1·02	·830
Flour of potatoes	·71	·790
Wood-ashes	·45	·484
Mahogany sawdust	·31	·524
Charcoal, powdered	·49	·637
“ “ and passed through silk	·41	·653
Coke, in powder	·77	1·290
TEXTILE MATERIALS.		
New calico of any density	....	·403
Cotton wool and sheep's wool	....	·323
Eider-down	....	·315
Hemp cloth, new	·54	·420
“ “ old	·58	·347
White writing-paper	·85	·347

Mr. Charles P. Paulding in "Steam in Covered and Bare Pipes" gives the following values from tests made by Professoı Jacobus and Mr. George M. Brill:

Material.	B. T. U. per Square Foot per Hour per One Degree Difference.		Specific Gravity.
	Jacobus.	Brill.	
Hair-felt	·32	·40	·207
Champion mineral wool	...	·47	·194
Mineral wool	...	·38	·266
Rock wool	...	·40	·248

In some instances compartments have been constructed of various types of insulation and tested for heat-losses.

In a paper by Mr. John E. Starr on "Insulation" before the "American Warehousemen's Association," October 1901, the results tabulated on page 295 are given.

With this experimental data at hand it is possible to obtain a logical basis upon which to compare the relative efficiencies of insulation. Theoretically we have to deal with the transmission of heat from without through walls which may be composed of brick, stone, wood, glass, or several of these in different combinations, or with air-spaces, to a mass of cold air on the inside of the compartment.

Prof. J. H. Kinealy, in his work translated from the German, "Formulas and Tables for Heating," gives the method adopted by German engineers, for obtaining the heat loss through various types of construction, which we adopt below.

If we assume a simple construction such as sheathing—insulation filling and sheathing as in Fig. 228—we may proceed as follows:

Let t_1 be the temperature of the air on A the warm side, t_6 the temperature on E the cold side, t_2, t_3, t_4, t_5 the temperatures of the faces of the insulation as shown; x and x_2 the thickness of the sheathing and x_1 that of the insulation filling in inches; c and c_2 the relative conductivities of the

TYPES OF INSULATION.

Materials.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.
MILL SHAVINGS.	
⅞" sheathing, paper, ⅞" sheathing, 8" mill shavings, ⅞" sheathing, paper, ⅞" sheathing	·056
Same slightly moist	·075
Same damp	·087
HAIR FELT.	
⅞" sheathing, paper, ⅞" sheathing, 1" hair felt, ⅞" sheathing, paper, ⅞" sheathing	·138
⅞" sheathing, ¼" hair felt, ⅞" sheathing, 2 layers ¼" hair felt, ⅞" sheathing, ¼" hair felt, ⅞" sheathing	·105
SHEET CORK.	
⅞" sheathing, paper, ⅞" sheathing, 1" air-space, 4" sheet cork, paper, ⅞" sheathing	·050
Same with 5" sheet cork	·037
⅞" sheathing, paper, 3" sheet cork, paper, ⅞" sheathing	·087
⅞" sheathing, paper, ⅞" sheathing, 1" sheet cork, ⅞" sheathing, paper, ⅞" sheathing	·137
GRANULATED CORK.	
⅞" sheathing, paper, ⅞" sheathing, 4" granulated cork, ⅞" sheathing, paper, ⅞" sheathing	·071
MINERAL WOOL.	
⅞" sheathing, paper, 2½" mineral wool, paper, ⅞" sheathing	·151
⅞" sheathing, paper, 1" mineral wool, paper, ⅞" sheathing	·192
WOOD.	
⅞" sheathing, paper, ⅞" sheathing, paper, ⅞" sheathing, paper, ⅞" sheathing	·178
AIR-SPACES.	
⅞" sheathing, paper, ⅞" sheathing, 8" air-space	·112
⅞" " " ⅞" " 8" "	
⅞" " " ⅞" " 8" "	
and ⅞" sheathing, paper, ⅞" sheathing	
PITCH.	
⅞" sheathing, 1" pitch, ⅞" sheathing	·204
⅞" " 2" " ⅞" "	·177
PUMICE, CALCINED.	
⅞" sheathing, paper, 2" pumice, paper, ⅞" sheathing	·140

sheathing *B* and *D*, and c_1 that of the insulation filling *C*. The flow of heat is by radiation and convection from the warm bodies and air to the outer surface of *B*, then by conduction through the insulation particle by particle, and finally by radiation and convection from the outer surface of *D*.

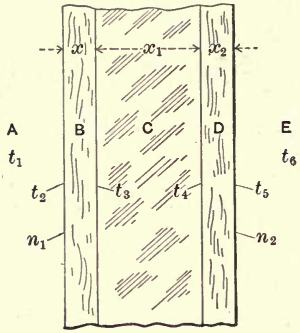


FIG. 228.

After the flow of heat through the wall has become stable, the amount of heat in British Thermal Units per hour passing any vertical unit section is the same as that passing any other equal parallel section, and any heat passing in from the *A* side passes out at the *E* side.

For clearness let us consider a vertical wall surface of one (1) sq. ft. area.

It has been determined by experiment that the amount of heat which will pass per hour from the warm air to this unit section is equal to the difference between the temperature of the air and that of the surface of the wall multiplied by a factor which depends upon the nature of the material composing the wall and the velocity of the air in contact with it. From this it follows that, if a represents this factor, the quantity of heat passing to this unit section of the insulation per hour is

$$a(t_1-t_2).$$

It has also been demonstrated by experiment that the amount of heat which will pass through a substance whose thickness is x and which has a coefficient of conductivity c will be equal to the difference in temperature of its two faces multiplied by $\frac{c}{x}$, or the amount of heat which will pass through the first sheathing B will be

$$\frac{c}{x}(t_2-t_3),$$

and similarly through the insulation filling C,

$$\frac{c_1}{x_1}(t_3-t_4),$$

and similarly through the sheathing D,

$$\frac{c_2}{x_2}(t_4-t_5).$$

The heat passing from the surface of D whose temperature is t_5 to the cool air E of temperature t_6 will be

$$a_1(t_5-t_6).$$

Let us assume that the total heat passing from the air A of a temperature t_1, through a unit section of the wall to the air E whose temperature is t_6, is equal to (t_1-t_6) times a factor which we will call n.

$$n(t_1-t_6).$$

The same amount of heat passes into and through the insulation as passes out through the last surface. Therefore

$$n(t_1-t_6)=\frac{c}{x}(t_2-t_3)=\frac{c_1}{x_1}(t_3-t_4)$$
$$=\frac{c_2}{x_2}(t_4-t_5)=a(t_1-t_2)=a_1(t_5-t_6)$$

or

$$\left.\begin{aligned} n(t_1-t_6)&=\frac{c}{x}(t_2-t_3)\\ &=\frac{c_1}{x_1}(t_3-t_4)\\ &=\frac{c_2}{x_2}(t_4-t_5)\\ &=a(t_1-t_2)\\ &=a_1(t_5-t_6)\end{aligned}\right\}\text{ and }\left\{\begin{aligned} \frac{nx}{c}(t_1-t_6)&=(t_2-t_3)\\ \frac{nx_1}{c_1}(t_1-t_6)&=(t_3-t_4)\\ \frac{nx_2}{c_2}(t_1-t_6)&=(t_4-t_5)\\ \frac{n}{a}\ (t_1-t_6)&=(t_1-t_2)\\ \frac{n}{a_1}\ (t_1-t_6)&=(t_5-t_6)\end{aligned}\right.$$

Adding, we get

$$(t_1-t_6)\left(\frac{nx}{c}+\frac{nx_1}{c_1}+\frac{nx_2}{c_2}+\frac{n}{a}+\frac{n}{a_1}\right)=t_1-t_6$$

or divided by (t_1-t_6) and transposing

$$n=\frac{1}{\dfrac{1}{a}+\dfrac{1}{a_1}+\dfrac{x}{c}+\dfrac{x_1}{c_1}+\dfrac{x_2}{c_2}}.$$

In order to solve the equation and obtain the B.T.U. transmitted per square foot of surface per hour per one degree difference we must know also the values of a and a_1.

The form of equation by Grashof as given by Prof. Kinealy is

$$a \text{ or } a_1=o+p\,\frac{(40o+30p)t}{10{,}000}.$$

The factor o depending upon the velocity of the air past the surface has the following values as given by Prof. Kinealy and determined by Rietschel:

VALUES OF *O.*

	O.
Air at rest	0·82
Air in slow motion, in contact with windows	1·03
Air in quick motion, air outside of building	1·23

Values of p by the same authorities are:

VALUE OF p.

Substance.	p	Substance.	p
Brickwork	0·74	Iron, rusted	0·69
Mortar	0·74	Cast iron, new	0·65
Stone masonry	0·74	Sheet iron, polished	0·092
Plaster of Paris	0·74	" " usual condition	0·57
Wood	0·74	" " cov'd with lead	0·13
Paper	0·78	Brass, polished	0·053
Glass	0·60	Copper	0·033
" with wet surface	1·09	Tin	0·045
		Zinc	0·049

We may neglect the part of the equation in solving T with little appreciable effect on the result.

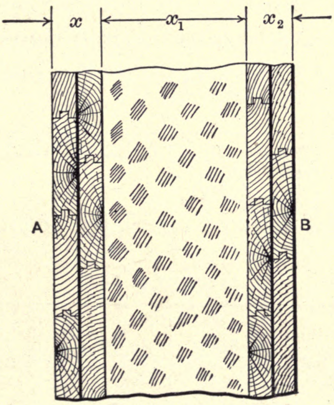


Fig. 229.—Type A.

For example, let us take type A insulation, Fig. 229, composed of $\frac{7}{8}$ in. sheathing, paper, $\frac{7}{8}$ in. sheathing, insulation filling, $\frac{7}{8}$ in. sheathing, paper and $\frac{7}{8}$ in. sheathing. We may assume that the paper acts only as a moisture retarder, then $x=1·75$ ins., $x_2=1·75$ ins., and x_1 = thickness of insulation filling, we have c and $c_2 = \cdot 75$ for relative conductivity of wood perpendicular to fibres. c_1 is the relative conductivity of the insulation filling.

If we assume the air on the warm side A of the insulation is in circulation and that on the cold side B is at rest, at the same time neglecting the factor involving T, we have

$$a = 1\cdot 23 + \cdot 74 = 1\cdot 97 \quad \frac{1}{a} = \cdot 508 \text{ for } A \text{ side,}$$

$$a_1 = \cdot 82 + \cdot 74 = 1\cdot 56 \quad \frac{1}{a_1} = \cdot 642 \text{ for } B \text{ side,}$$

and

$$n = \frac{1}{\cdot 508 + \cdot 642 + \frac{1\cdot 75}{\cdot 75} + \frac{x_1}{c_1} + \frac{1\cdot 75}{\cdot 75}} = \frac{1}{5\cdot 82 + \frac{x_1}{c_1}} \quad . \quad (a)$$

Unfortunately there are a great many materials suitable at present for insulating purposes which have not been sufficiently experimented upon to determine the constant c; we have some constants, however, and may arrive at others indirectly.

If we assume $c = \cdot 60$ for spruce mill shavings, a value which seems fair from the data at hand, we can reduce equation (a) to

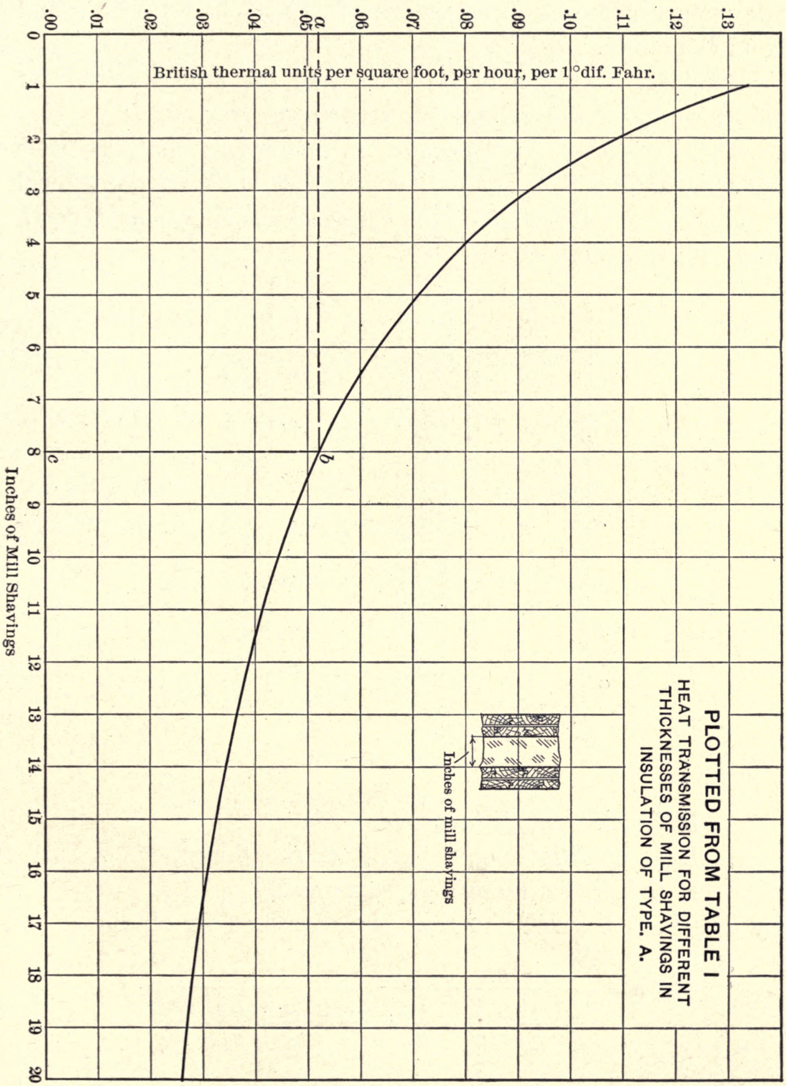
$$n = \frac{1}{5\cdot 82 + \frac{x_1}{\cdot 60}} = \frac{1}{5\cdot 82 + 1\cdot 67 x_1}.$$

Values for mill shavings as calculated from this formula are as tabulated:

TABLE I.

Inches of Mill Shavings x.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.	Inches of Mill Shavings x.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.	Inches of Mill Shavings x.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.
1 inch	·133	9 inches	·048	17 inches	·0292
2 inches	·109	10 "	·044	18 "	·0279
3 "	·092	11 "	·041	19 "	·0267
4 "	·080	12 "	·039	20 "	·0255
5 "	·071	13 "	·036	21 "	·0245
6 "	·063	14 "	·034	22 "	·0235
7 "	·057	15 "	·032	23 "	·0226
8 "	·052	16 "	·0308	24 "	·0218

The following curve is plotted from the constants in this table.



For example, referring to this curve, when x_1 in type A construction is equal to 8 inches, then the British Thermal Units which will pass through one square foot of wall surface for each degree difference between the two faces will be ·052 B.T.U.; the transmission for other values of x_1 may likewise be found from the curve.

Similarly values may be calculated for any material for which we know the value of c_1 and curves plotted.

The comparative efficiencies of insulations may be thus determined, and after estimating their cost, the most economical type selected. If we adopt the construction shown as type B, Fig. 230, composed of

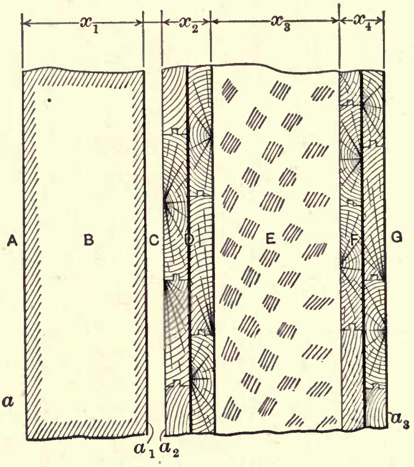


FIG. 230.—Type B.

B (brick wall), C (air-space),
D ($\frac{7}{8}$ in. sheathing, paper, $\frac{7}{8}$ in. sheathing),
E (insulation filling), F (same as D),

by a deduction similar to that just described we obtain

$$n=\frac{1}{\frac{1}{a}+\frac{1}{a_1}+\frac{1}{a_2}+\frac{1}{a_3}+\frac{x_1}{c_1}+\frac{x_2}{c_2}+\frac{x_3}{c_3}+\frac{x_4}{c_4}},$$

the factors a, a_1, a_2, and a_3 depending upon the velocity of the air by the surfaces shown, also the character of these surfaces and x_1, x_2, x_3, and x_4 representing the thicknesses of the various materials, while c_1, c_2, c_3, and c_4 represent their relative conductivities.

If we assume that the air is still in the air-space C and also at G, the cool side of the insulation, we will have

$$a_1 = a_2 = a_3 = \cdot 642.$$

If also we assume the air in circulation at A the warm side, we will have

$$a = \cdot 508.$$

Then

$$n = \frac{1}{\cdot 508 + 3(\cdot 642) + 2\left(\frac{1\cdot 75}{\cdot 75}\right) + \frac{x_1}{5\cdot 56} + \frac{x_3}{c_3}} = \frac{1}{7\cdot 1 + \cdot 18x + \frac{x_3}{c_3}}.$$

Substituting for x_1 different values corresponding to walls of various thicknesses we may reduce the formula to the following, assuming that the insulation is mill shavings and x is the thickness in inches of this filling:

For 4-in. brick wall $n = \dfrac{1}{7\cdot 82 + \dfrac{x}{\cdot 60}}$.

For 8-in. brick wall $n = \dfrac{1}{8\cdot 54 + \dfrac{x}{\cdot 60}}$.

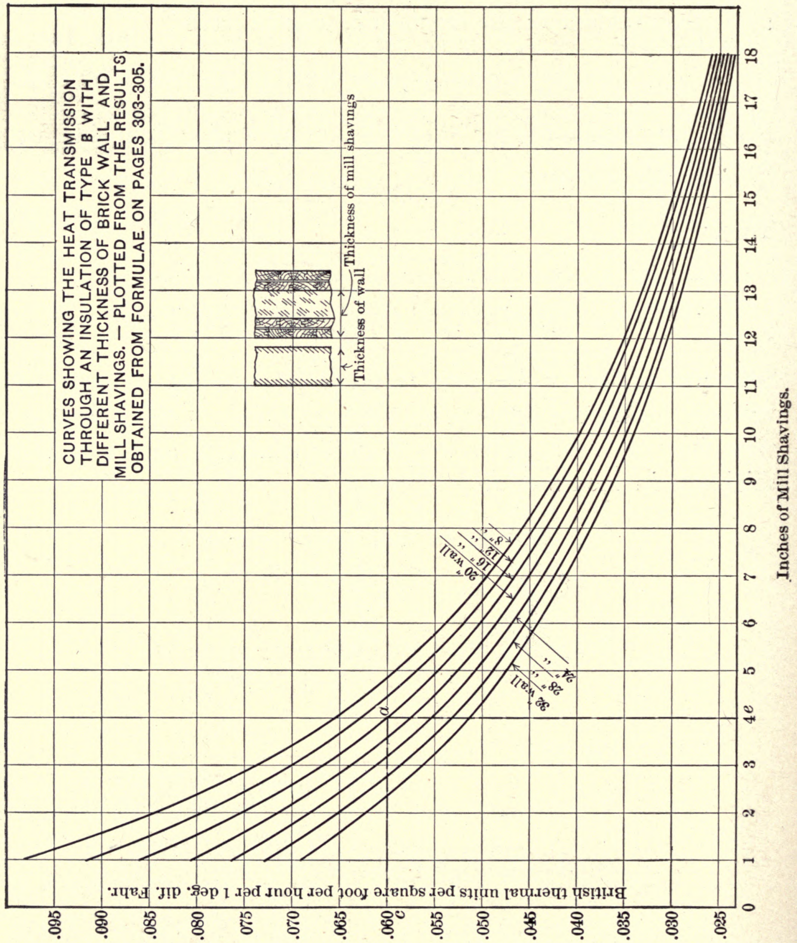
For 12-in. brick wall $n = \dfrac{1}{9\cdot 26 + \dfrac{x}{\cdot 60}}$.

For 16-in. brick wall $n = \dfrac{1}{9\cdot 98 + \dfrac{x}{\cdot 60}}$.

For 20-in. brick wall $n = \dfrac{1}{10\cdot 70 + \dfrac{x}{\cdot 60}}$.

For 24-in. brick wall $n = \dfrac{1}{11\cdot 42 + \dfrac{x}{\cdot 60}}$.

For 28-in. brick wall $n=\dfrac{1}{12\cdot13+\dfrac{x}{\cdot60}}$.



For 32-in. brick wall $n=\dfrac{1}{12\cdot86+\dfrac{x}{\cdot60}}$.

For 36-in. brick wall $n=\dfrac{1}{13\cdot58+\dfrac{x}{\cdot60}}$.

For 40-in. brick wall $n=\dfrac{1}{14\cdot30+\dfrac{x}{\cdot60}}$.

The curves from the results of these equations will be found on page 304 and the heat transmission for any combination of wall and insulation easily found.

For the construction shown as type C, Fig. 231, which is composed of B (sheathing, paper, sheathing), C (air-space), D (same as B), E (insulation filling), F (same as B), G (air-space), H (same as B), we shall have

$$n=\frac{1}{\dfrac{1}{a}+\dfrac{1}{a_1}+\dfrac{1}{a_2}+\dfrac{1}{a_3}+\dfrac{1}{a_4}+\dfrac{1}{a_5}+\dfrac{x}{c}+\dfrac{x_1}{c_1}+\dfrac{x_2}{c_2}+\dfrac{x_3}{c_3}+\dfrac{x_4}{c_4}},$$

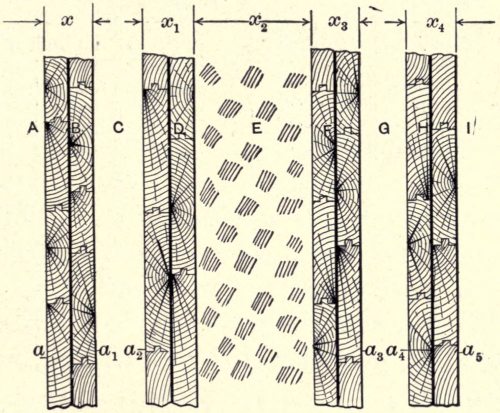


FIG. 231.—Type C.

the factors a, a_1, etc., depending upon the air velocity and the character of the surfaces, and x, x_1, etc., the thicknesses in inches of the materials making up the insulation, while c, c_1, etc., is the relative conductivity of each. If the air at *A* is in quick motion, and that at *C*, *G*, and *I* still, then $\dfrac{1}{a}=\cdot508$ and $\dfrac{1}{a_1}=\dfrac{1}{a_2}=\dfrac{1}{a_3}=\dfrac{1}{a_4}=\dfrac{1}{a_5}=\cdot642$, $x=x_1=x_3=x_4=1\cdot75$ inches and $c=c_1=c_3=c_4=\cdot75$.

$$n=\frac{1}{\cdot 508+5(\cdot 642)+4\left(\frac{1\cdot 75}{\cdot 75}\right)+\frac{x_2}{c_2}}=\frac{1}{13\cdot 06+\frac{x_2}{c_2}}.$$

When x_2 is the thickness of the shavings and c_2 their coefficient of conductivity, for an insulation filling of mill shavings we will have the following values, the equation reducing to

$$n=\frac{1}{13\cdot 06+1\cdot 67x}.$$

In Table IV are values based on this formula.

TABLE IV.

Inches of Shavings.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.	Inches of Shavings.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.	Inches of Shavings.	B. T. U. per Sq. Ft. per Hour per One Degree Difference.
1 inch	·067	7 inches	·041	13 inches	·0288
2 inches	·061	8 "	·038	14 "	·0274
3 "	·055	9 "	·035	15 "	·0263
4 "	·051	10 "	·0335	16 "	·0262
5 "	·046	11 "	·0318	17 "	·0242
6 "	·043	12 "	·0302	18 "	·0229

The curve for these constants appears on the following page. While we have deducted constants and plotted only curves for insulation which consists of mill shavings in combination with wood, brick, and air-spaces, similar ones could easily be constructed for other material such as cork, mineral wool, hair, felt, etc.

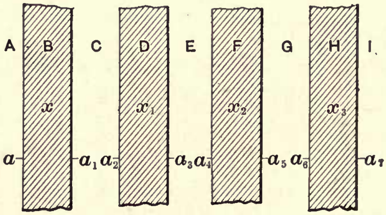
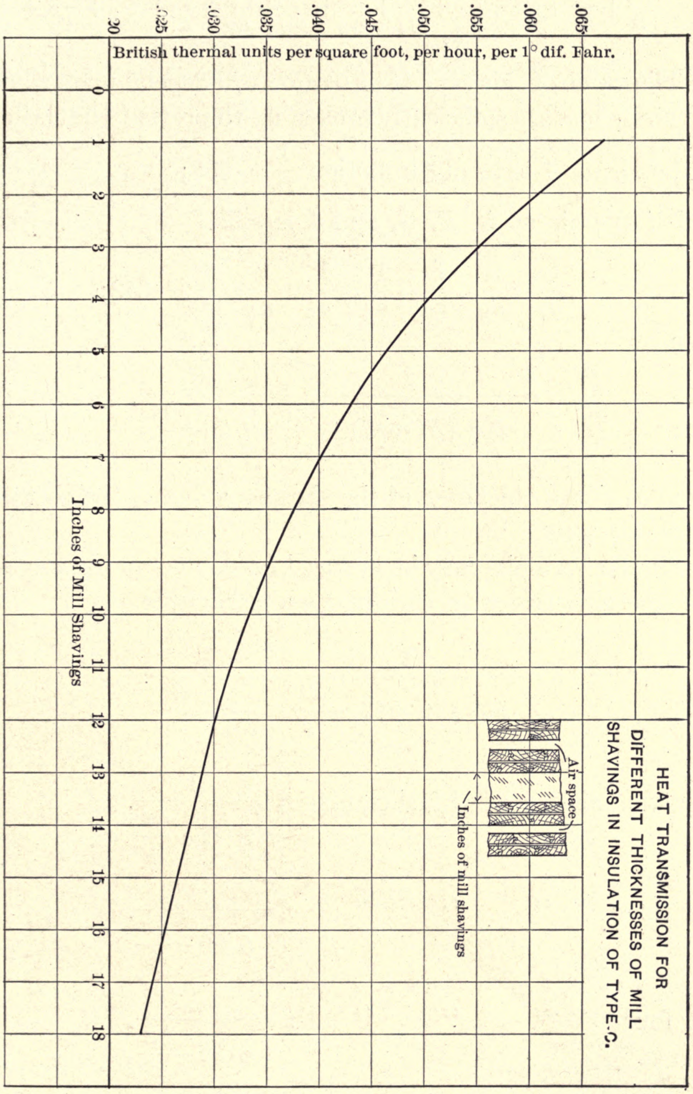


FIG. 232.—Type D.

Enough, with the following additional example of a series of air-spaces, has been shown to illustrate the method adopted.

For the construction shown as type D, Fig. 232, which is



HEAT TRANSMISSION FOR DIFFERENT THICKNESSES OF MILL SHAVINGS IN INSULATION OF TYPE C.

composed of B (single or double sheathing or any other material), C (air-space), D (same as B, etc.).

$$n=\frac{1}{\frac{1}{a}+\frac{1}{a_1}+\frac{1}{a_2}+\frac{1}{a_3}+\frac{1}{a_4}+\frac{1}{a_5}+\frac{1}{a_6}+\frac{1}{a_7}+\frac{x}{c}+\frac{x_1}{c_1}+\frac{x_2}{c_2}+\frac{x_3}{c_3}}.$$

The value of his type of insulation will depend upon whether the air-spaces are sufficiently broken up to prevent circulation. If the air at *A* is in quick motion $\frac{1}{a}=\cdot508$;

and if the air at *C, E, G,* and *I* is still

$$\frac{1}{a_1}=\frac{1}{a_2}=\frac{1}{a_3}=\frac{1}{a_4}=\frac{1}{a_5}=\frac{1}{a_6}=\frac{1}{a_7}=\cdot642;$$

then

$$n=\frac{1}{\cdot508+7(\cdot642)+\frac{4x_1}{c_1}}=\frac{1}{5\cdot00+\frac{4x_1}{c_1}},$$

when x_1 is the thickness of the sheathing and c_1 the coefficient of conductivity.

Similar formulæ may be deduced for one or more air-spaces: For one air-space with sheathing on either side

$$n=\frac{1}{\cdot508+3(\cdot642)+\frac{2x}{c}}=\frac{1}{2\cdot43+\frac{2x}{c}}.$$

For two air-spaces with sheathing $n=\frac{1}{3\cdot72+\frac{3x}{c}}$.

For three " " " $n=\frac{1}{5\cdot00+\frac{4x}{c}}$.

For four " " " $n=\frac{1}{6\cdot29+\frac{5x}{c}}$.

For five " " " $n=\frac{1}{7\cdot57+\frac{6x}{c}}$.

For six air-spaces with sheathing $n=\dfrac{1}{8\cdot85+\dfrac{7x}{c}}$.

For the transmission of heat through walls, in British Thermal Units per square foot per hour per one degree Fahr. difference, the values used by the state of Prussia, Germany, as given by Mr. J. H. Kinealy are:

FOR BRICKWORK.

Inches Thick.	B. T. U.
4·72	0·492
9·85	0·348
15·0	0·266
20·1	0·226
25·2	0·184
30·3	0·164
35·4	0·133
40·5	0·123
45·6	0·113

FOR SANDSTONE MASONRY, BLOCK OR RUBBLE.

Inches Thick.	B. T. U.
11·8	0·451
15·7	0·390
19·7	0·348
23·8	0·318
27·6	0·287
31·5	0·266
35·4	0·246
39·4	0·226
43·3	0·205
47·2	0·195

	B. T. U.
Plaster from 1·6 to 2·6 inches thick on wire lathing....	0·615
“ “ 2·6 “ 3·2 “ “ “ “ “	0·492

WINDOWS AND SKYLIGHTS.

	B. T. U.
Single window..................................	1·09
“ “ double glass..........................	0·62
Double window..................................	0·46
Single skylight..................................	1·16
Double “	0·48

We have computed the following constants for brick walls, usually found in American practice.

BRICKWORK.

Inches Thick.	B. T. U.
4	0·437
8	0·388
12	0·304
16	0·249
20	0·212
24	0·184
28	0·163
32	0·146
36	0·132
40	0·120

In deductions relative to heat losses, some assumption is necessarily made; rigid exactness is neither possible nor desirable, for all computations are based on conditions which are constantly changing and upon quantities which are never more than approximately correct. A change in the velocity of the air has a marked effect on the amount of heat carried away from any surface; in fact, the varying conditions even in a short period would present insurmountable obstacles to rigid exactness.

All calculations may be considered as approximate for the guidance of the engineer whose reputation is due not alone to his knowledge in the abstract but also to his ability to judge conditions and to make suitable allowances. A refrigerating apparatus designed for any purpose should have sufficient reserve capacity to easily take care of any contingency which may arise.

CHAPTER IV.

ICE-MAKING.

WHILE there are several methods for making ice, such as the "Vacuum," the "Holden," and other processes, those extensively used are the can and the plate systems.

CAN ICE-MAKING.

1. **General Method.**—Primarily this method is one in which cans filled with water are immersed in a freezing solution until the contents are frozen into blocks varying in size and weight. Fig. 233 is a sectional elevation of a typical American can ice-tank. The freezing solution surrounding the cans is usually calcium chloride brine, and may be cooled either by direct expansion or brine coils, with propellor agitators to keep the surrounding solution in motion. The brine may also be cooled by a brine cooler and circulated through the ice-tank.

The type of ice-can usually adopted by American ice-machine builders and the approximate time required to freeze ice with a brine bath of 15° F. is as tabulated.

2. **Size of Cans.**—These dimensions are often varied to suit conditions.

Weight of Cake.	Inside Dimensions of Cans.			Thickness of Metal, U. S. Gauge.	Time Required to Freeze.
	Top.	Bottom.	Length Inside.		
50 lbs.	8″×8″	7½″×7½″	31 inches	No. 16	20 hours
100 "	8″×16″	7¼″×15¼″	31 "	" 16	36 "
200 "	11½″×22½″	10½″×21½″	31 "	" 16	55 "
300 "	11½″×22½′	10½″×21½″	44 "	" 16	60 "
400 "	11½″×22½″	10½″×21½″	57 "	" 16	60 "

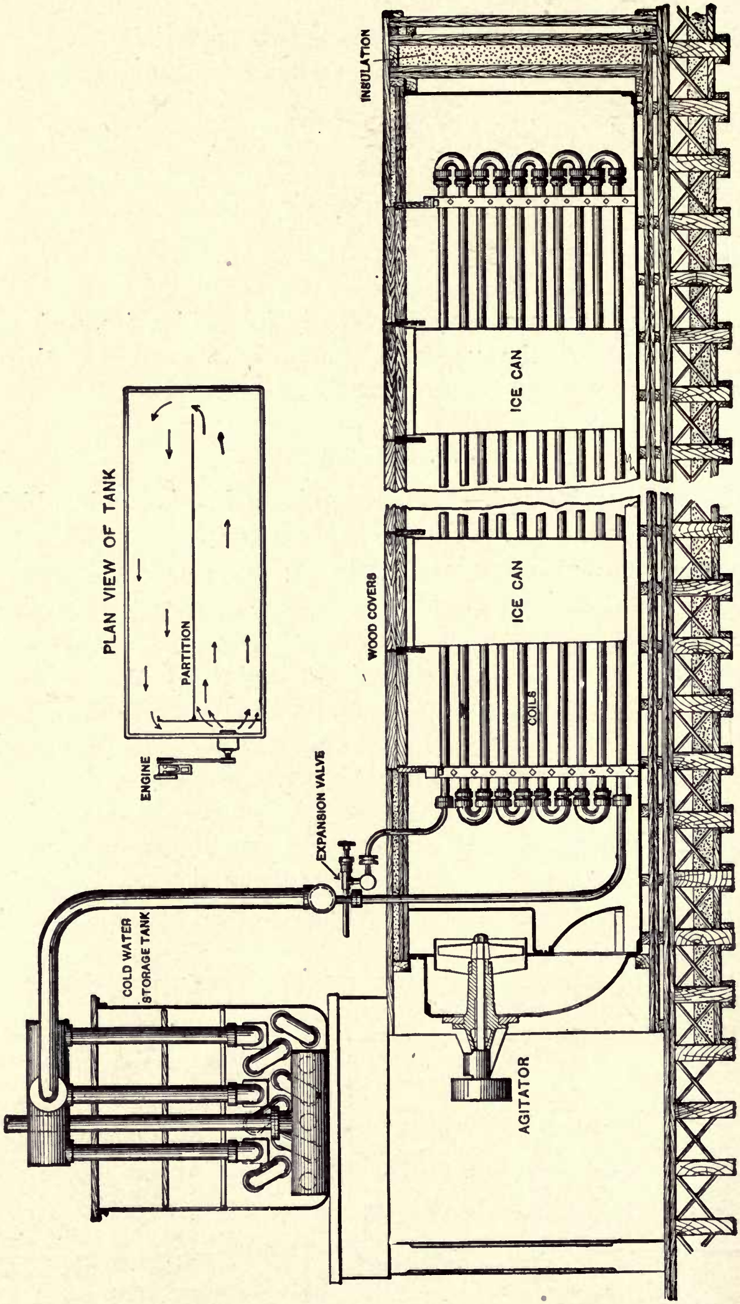


FIG. 233.—Can Ice-tank.

FRICK COMPANY.

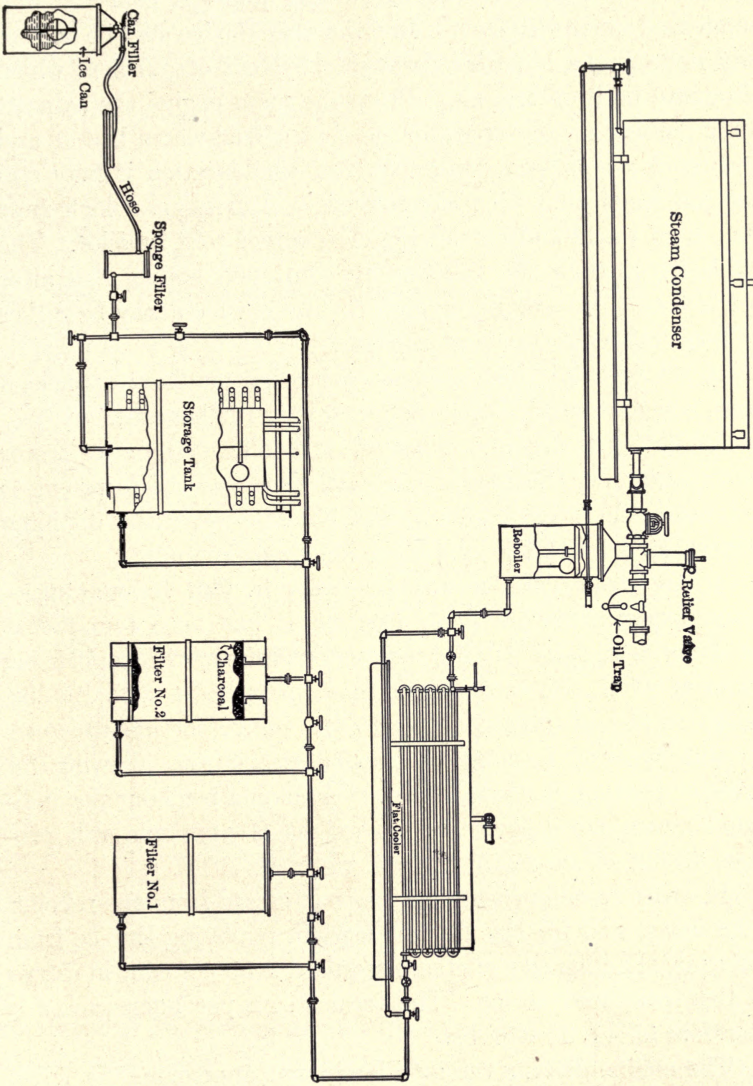


FIG. 234.—Distilling System.

TRIUMPH ICE MACHINE COMPANY.

3. Purifying Water for Ice-cans.—All the steam which is used for the power of the ice-making plant is collected and condensed for the ice-cans. In order that the ice may be neither oily nor opaque it passes through a system of purifiers which eliminate both oil and air. From the main engine the exhaust after passing an oil-separator goes to a feed-water heater and thence to a steam-condenser. After condensation it may still contain some oil. This is removed in a skimmer which runs the oil to waste and carries the clear water to a reboiler. The air is expelled in this reboiler by continued boiling and after being forecooled the water goes to filters, which may be either of bone charcoal, sand, or sponges.

A storage-tank, containing either brine or direct-expansion coils, cools the water before deliverig it to the ice-cans.

While there are numerous other similar systems of water purification for the manufacture of clear ice, the foregoing is a typical American method. Fig. 234 illustrates the distilling system used by the Triumph Ice Machine Company.

4. American Practice and Economy in Can Ice-making.—From experiments by Prof. Denton, it has been found that for American practice under summer conditions, with condensing water at 70° F., the compression machine operates with 190 lbs. gauge condenser pressure and 15 lbs. gauge suction pressure. In this type of machine the useful circulation, allowing for cylinder heating, is about 13 lbs. of ammonia per hour per indicated steam horse-power. This weight of ammonia will produce 32 lbs. of ice at 15° F. from water at 70° F.

In order to compensate for losses of steam from the reboiler from leaks, and for the amount used in removing the ice from the cans, the quantity of steam required is 33 per cent in excess of that produced as ice. The total steam per horse-power is therefore $32\times1\cdot33=43$ lbs.

The engine driving the axuiliaries uses approximately 7 lbs. of this; the steam consumption of the main engine driving the compressor is therefore 36 lbs. per indicated horse-power per hour.

A more economical engine is usually inadvisable since live steam would be needed to make up the difference between the steam used and the amount needed for the ice-cans. If we assume an evaporation of 8½ lbs. of water per pound of coal, the amount of ice per one pound of coal would be 6 lbs., which is the average result in American practice. Under some conditions compound condensing engines are economically installed. In order to profitably use such an engine in can ice-making a re-evaporator is used, such as is shown in Fig. 235.

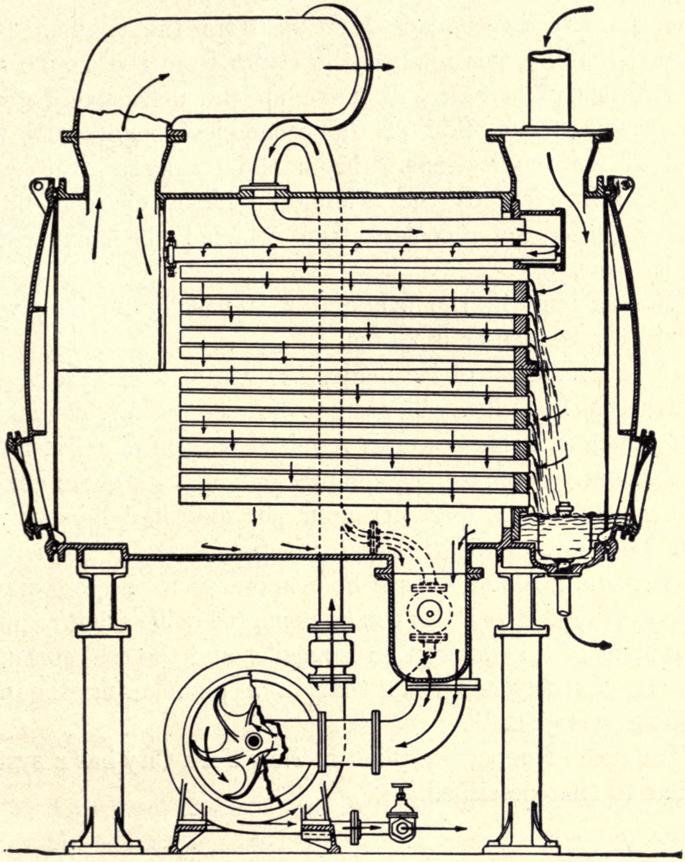


FIG. 235. "LILLIE" EVAPORATOR.

In operation the exhaust-steam from the engine enters the tubes of the re-evaporator and is condensed by the cooling-water dripping over the outside; the cooling-water in turn is partially evaporated by the heat from the steam and goes as a vapor to a condenser, where it is condensed and collected with the already condensed steam from the engine. It is claimed that by this method the steam from the engine may be sufficiently increased to make up the deficiency between the steam consumption of the economical engine and that required for the ice-cans. From the condenser, the water goes consecutively to the skimmer, reboiler, charcoal filters, and condensed-water coolers, from which it is finally fed to the ice-cans.

The vacuum in the evaporator is usually 20 ins., while that in the condenser is 25 ins.

If for every ton of ice made, 400 lbs. of water are wasted in leakages, 2400 lbs. will need to be distilled for ice-making. If 3½ indicated horse-power is required per ton of refrigeration in 24 hours to operate the compressor and with these engines 17 lbs. of steam is used per hour per indicated horse-power, then 1400 lbs. of distilled water is furnished by the engine. The remaining 1000 lbs. must be evaporated in the evaporator.

It is evident that if this can be done, the boiler plant required for a plant of this description is smaller and the coal consumption consequently less than that in a plant using the non-condensing-engine.

The Jacob Ruppert ice-plant in New York City has a system similar to that described.

CHAPTER V.

PLATE ICE-MAKING.

1. General Method.—In this method, plates in which the freezing solution circulates are immersed in a tank containing water. The plates are usually a series of horizontal pipes near enough together to form a vertical surface. The water next to this surface is frozen, forming a wall on either side which increases in thickness until a sufficiently heavy plate of ice is obtained.

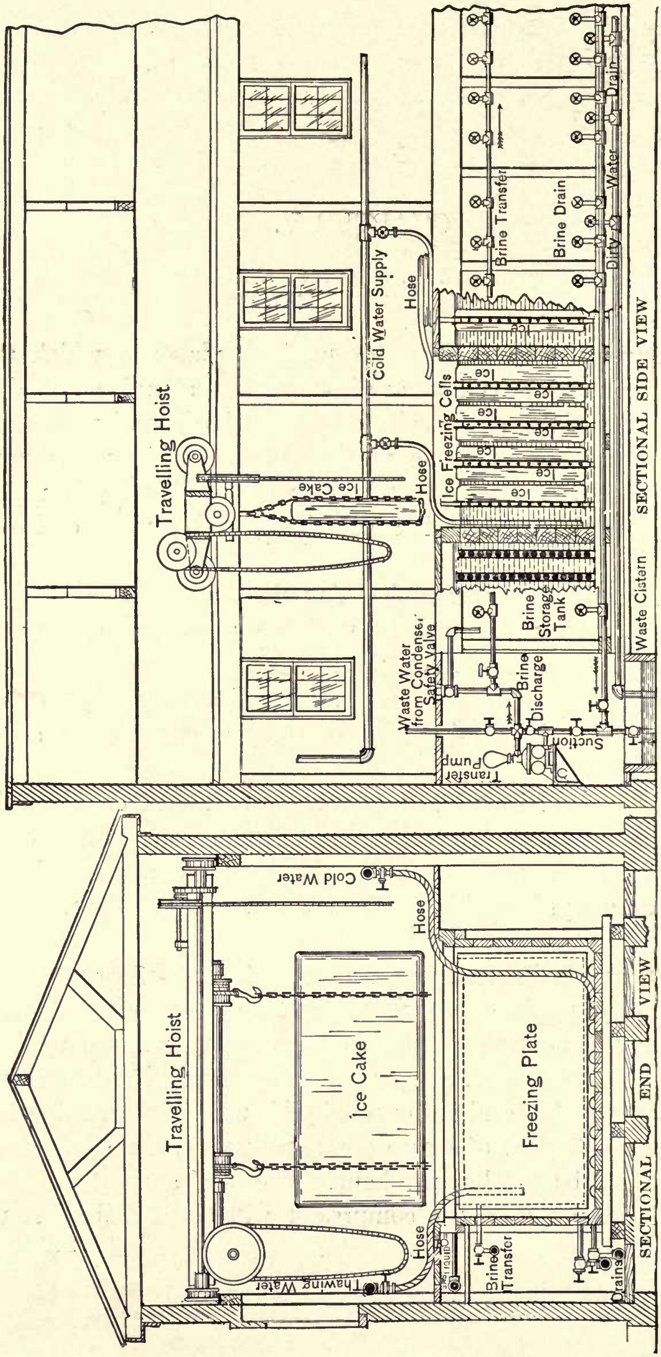
Fig. 236 illustrates a typical American plate ice-tank.

Any potable water may be used for plate ice-making provided it has first been properly filtered.

To insure clear ice free from air-bubbles the water is either kept in motion by the injection of compressed air or by a circulating pump.

The plates of ice when frozen are separated from the coils by passing hot gas or solution through these or separate coils, afterward being removed from the tank by a crane hoist, placed on a tipping table, and sawed into cakes of the requisite size.

2. American Practice and Economy in Plate Ice-making.—In ordinary practice economical steam-engines using 14 lbs. of steam per hour per indicated horse-power may be used to drive the compressor. From tests by Prof. Denton about 15 per cent of the total power required is used by the auxiliaries. If these use 50 lbs. of steam per horse-power, the steam used by them will be 7·5 lbs. per hour. The total amount used per steam horse-power of the compressor will be 21·5 lbs. If we



FRICK COMPANY.

FIG. 236.—Plate Ice-tank.

assume an evaporation of 8 lbs. of water per pound of coal, we will have 11·8 lbs. of ice per pound of coal.

The best result thus far in American practice for this type of installation is 10·5 lbs. of ice per pound of coal, which would be the result with an engine using 17 lbs. of steam per hour per I.H.P. In a recent installation, however, a departure has been made by using an absorption machine operated with exhaust-steam. The engine which furnished power for the auxiliaries has a back pressure of 33 lbs. absolute and exhausts through a superheater to the generator steam-coils, furnishing all the steam required.

During a test run of seven days, from apparently reliable data, the plant produced 14·04 lbs. of ice per one pound of coal.

CHAPTER VI.

REFRIGERATING MACHINERY.

1. The Unit of Refrigeration usually adopted in designating the size of refrigerating machinery is the refrigerating capacity of one ton (2000 lbs.) of ice melting in 24 hours, which is equivalent to 284,000 British Thermal Units in 24 hours, 11,833·3 B.T.U. in 1 hour, or to 197·2 B.T.U. per minute.

Refrigerating machines are sometimes rated upon the number of gallons of calcium-chloride brine solution of a given strength which they can cool per minute from one temperature to another. This is a preferable commercial standard since each machine is rated according to the work which it can perform under actual conditions. A commission was recently appointed by the A. S. M. E. to investigate and establish a satisfactory standard unit of refrigeration acceptable to American ice-machine builders. Some investigation and progress has thus far been made.

The ice-making capacity of refrigerating machinery is usually taken as 60 per cent of its refrigerating capacity.

2. General Types of Refrigerating Machines.—*Ammonia Refrigerating Machines* are of two distinct types, the compression and the absorption, differing in construction and operation. The compression type used in American practice is similar in many ways to that used in Germany. It is constructed either with single- or double-acting ammonia compression cylinders on vertical or horizontal frames and may be operated either by simple non-condensing or compound-condensing engines. The compressors are operated both under the wet and the dry compression system.

3. In wet compression some of the ammonia enters the compressor cylinders in a liquid state, the heat developed during compression being used up in converting the liquid into vapour, and consequently there is a saturated vapour at the end of compression which has a boiling-point corresponding to the condenser pressure. In this type no water-jacket is required around the ammonia cylinder.

4. In dry compression the ammonia entering the compressor cylinder is all in a gaseous condition, so that the heat developed during compression, if no water-jacket is used, superheats the gas several hundred degrees above the temperature corresponding to the condenser pressure. A water-jacket surrounds the ammonia cylinder, however, and absorbs the heat, permitting cylinder lubrication.

5. **Economy of Compression Machines.**—From experiments made upon American compression machines it has been found that their capacities are proportional almost entirely to the weight of gas discharged. This weight depends upon the suction pressure and the actual cubic displacement of the compressor.

The economy in coal consumption depends mainly upon the suction and condenser pressures. The following table, from " Siebel's Compend of Mechanical Refrigeration," gives the number of cubic feet of gas which must be pumped per minute at different suction and condenser pressures to produce one ton of refrigeration in 24 hours.

If we assume that the quantity of gas discharged is equal to the displacement of the piston in cubic feet times the weight of one cubic foot of gas at the suction pressure, we are misled. From experiments upon compression machines operated under both the wet and dry systems, where the ammonia has been metered, it has been found that approximately 25 per cent less weight of gas passes the compressor than is accounted for this by assumption. In dry compression the loss in displacement is probably due to the superheating effect which the metal surfaces of the inlet valves and openings have upon the entering gas, rarefying it and reducing its weight correspondingly.

CUBIC FEET OF AMMONIA-GAS THAT MUST BE DISCHARGED PER MINUTE AT DIFFERENT CONDENSER AND SUCTION PRESSURES TO PRODUCE ONE TON OF REFRIGERATION IN TWENTY-FOUR HOURS.

Temperature of Gas in Degrees F.	Corresponding Suction Pressure, Pounds per Sq. In., Gauge Pressure.	Temperatures of Gas in Degrees F.								
		65°	70°	75°	80°	85°	90°	95°	100°	105°
		Corresponding Condenser Pressures (gauge), Pounds per Sq. In.								
		103°	115°	127°	139°	153°	168°	184°	200°	218°
−27	1	7·22	7·30	7·37	7·46	7·54	7·62	7·70	7·79	7·86
−20	4	5·84	5·90	5·96	6·03	6·09	6·16	6·23	6·30	6·43
−15	6	5·35	5·40	5·46	5·52	5·58	5·64	5·70	5·77	5·83
−10	9	4·66	4·73	4·76	4·81	4·86	4·91	4·97	5·05	5·08
− 5	13	4·09	4·12	4·17	4·21	4·25	4·30	4·35	4·40	4·44
0	16	3·59	3·63	3·66	3·70	3·74	3·78	3·83	3·87	3·91
5	20	3·20	3·24	3·27	3·30	3·34	3·38	3·41	3·45	3·49
10	24	2·87	2·90	2·93	2·96	2·99	3·02	3·06	3·09	3·12
15	28	2·59	2·61	2·65	2·68	2·71	2·73	2·76	2·80	2·82
20	33	2·31	2·34	2·36	2·38	2·41	2·44	2·46	2·49	2·51
25	39	2·06	2·08	2·10	2·12	2·15	2·17	2·20	2·22	2·24
30	45	1·85	1·87	1·89	1·91	1·93	1·95	1·97	2·00	2·01
35	51	1·70	1·72	1·74	1·76	1·77	1·79	1·81	1·83	1·85

In wet compression this action apparently does not take place to such an extent, but the gas in the clearance space, which is not expelled at the condenser pressure, expands as the piston recedes and excludes part of the new charge, thus reducing the capacity of the machine. The clearance in single-acting compressors, which are usually operated under dry compression, is as low as $\frac{1}{3}$ of 1 per cent, while double-acting compressors, which are usually operated under wet compression, have clearances several times greater. In using the preceding table allowances must be made for this loss of displacement.

6. Types of Compression Machines.—The several types given in the following illustrations have been designed and constructed by American builders, and successfully operated in plants throughout the United States.

A single-acting compressor of the "York" pattern, as in Fig. 237, also in section, as in Fig. 238, is operated by a simple. non-condensing engine. Fig. 239 is a sectional cut of a "Frick" compression cylinder of the single-acting type, showing the arrangement of the suction- and discharge-valves and ports. In these the rod packings are never subjected to more than the suction pressure.

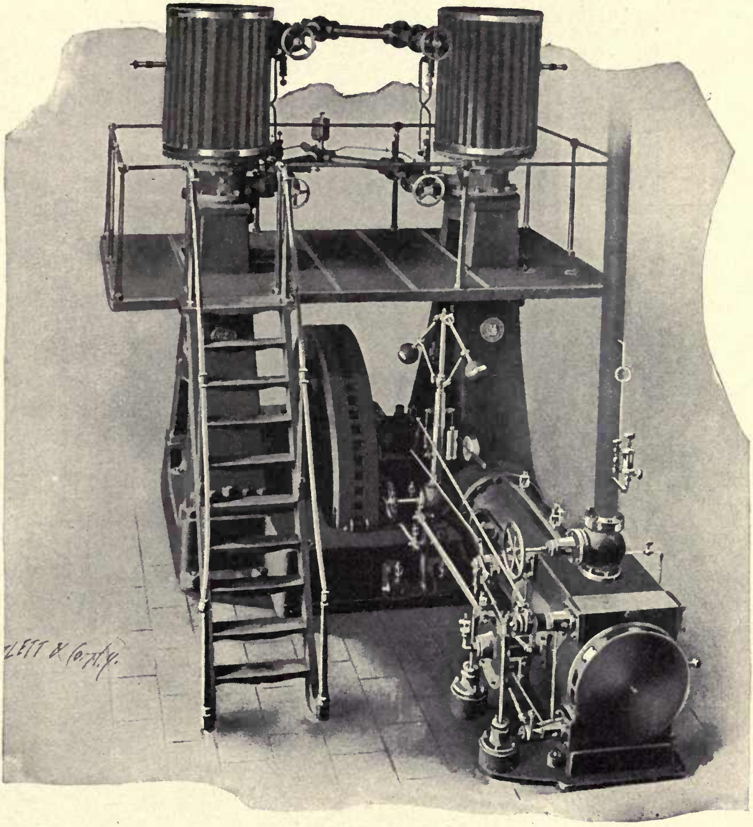


FIG. 237. YORK MFG. CO.

The ammonia compressor may be operated also, as in the "York" single-acting compressor, Fig. 240 and Fig. 241, by a tandem compound engine, or as in the double-acting compressor of the "De La Vergne" type, Fig. 242, by a cross-compound engine.

Fig. 243 is a sectional cut of the ammonia cylinder for this type. In some cases oil is injected into the cylinder to cool the compressed gas and also to seal the rod, eliminating excessive leakage of ammonia through the packing. This is illustrated by Fig. 247. A horizontal compressor of the "De La Vergne"

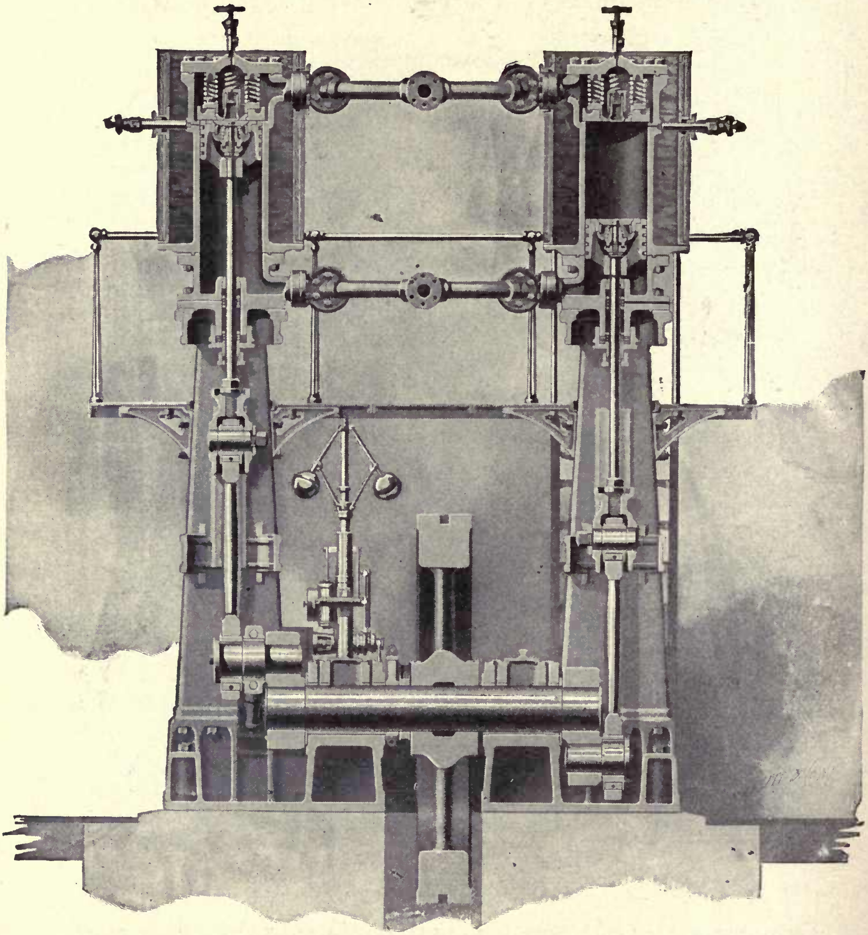


FIG. 238. YORK MFG. CO.

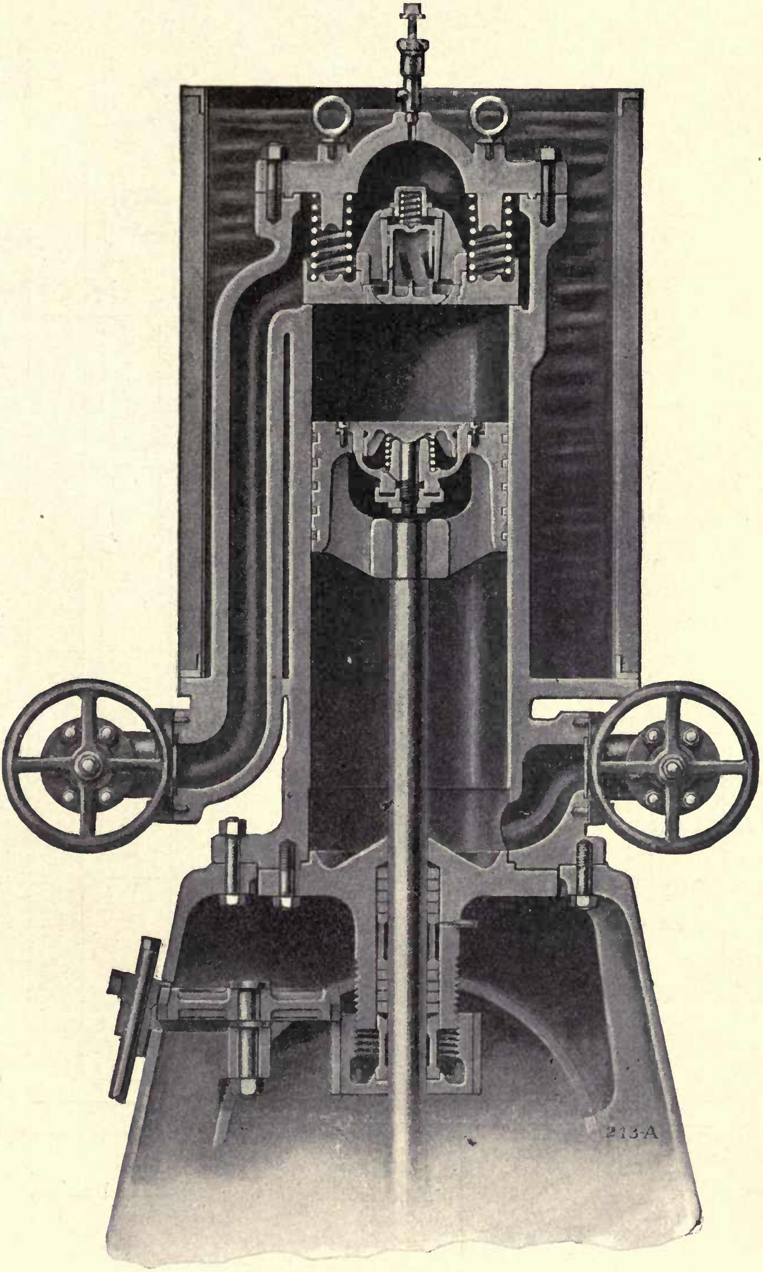


Fig. 239 **Frick Company.**

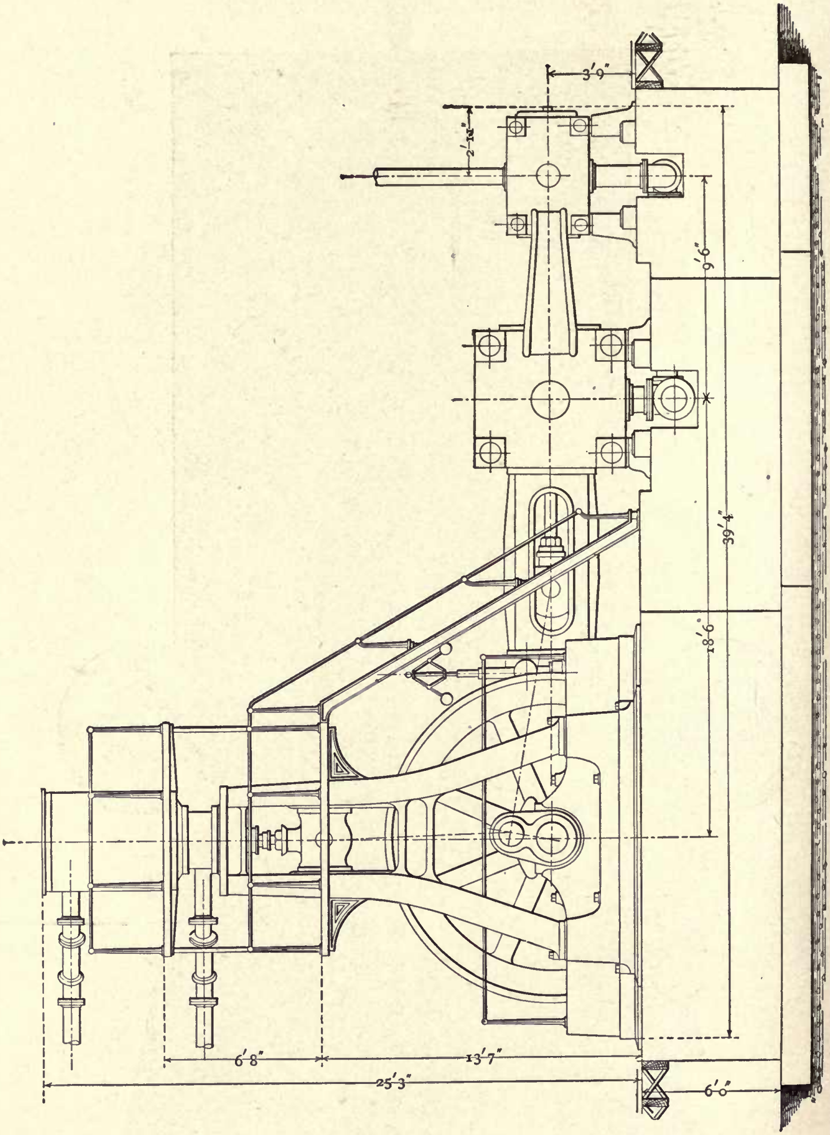


FIG. 240.

YORK MFG. CO.

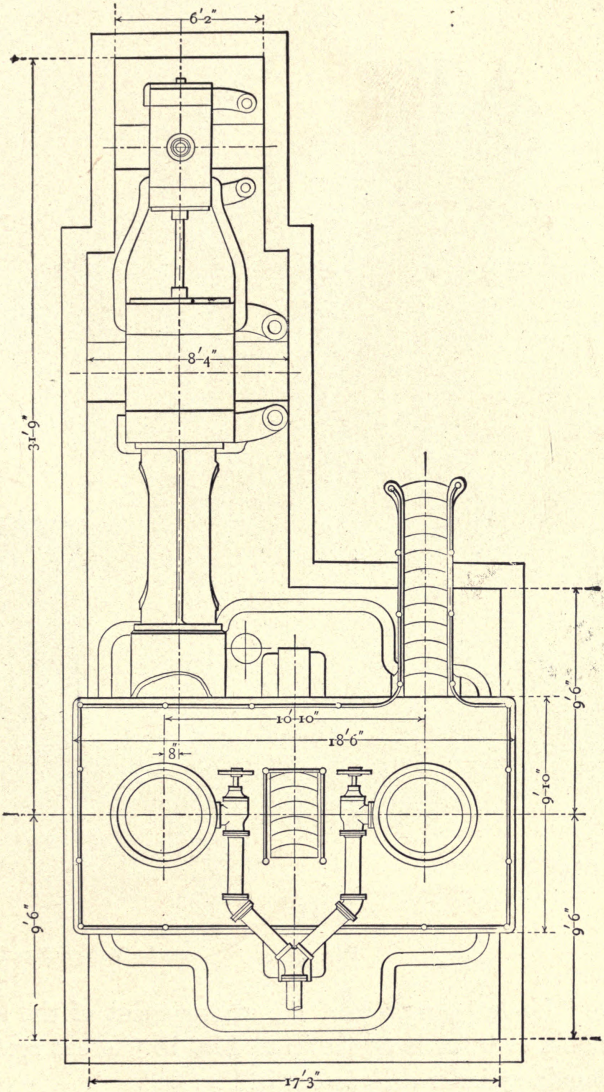


FIG. 241.

YORK MFG. CO.

type, as shown in Fig. 245, has ammonia cylinders, as in Fig. 246.

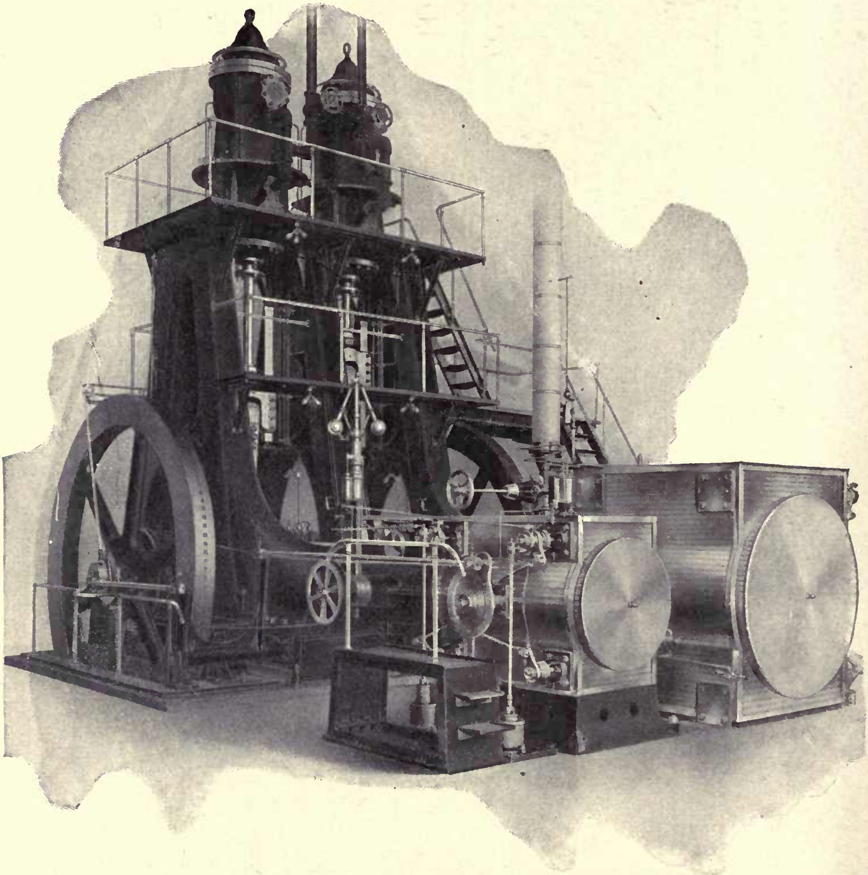


FIG. 242. DE LA VERGNE MACHINE CO.

In the "Linde" compressor, the arrangement of the steam and ammonia cylinder is as shown in Fig. 18 (Lorenz section), with ammonia cylinder designed as in Fig. 248.

There are a great many other machines of American manufacture differing but little in general design from these illustrated.

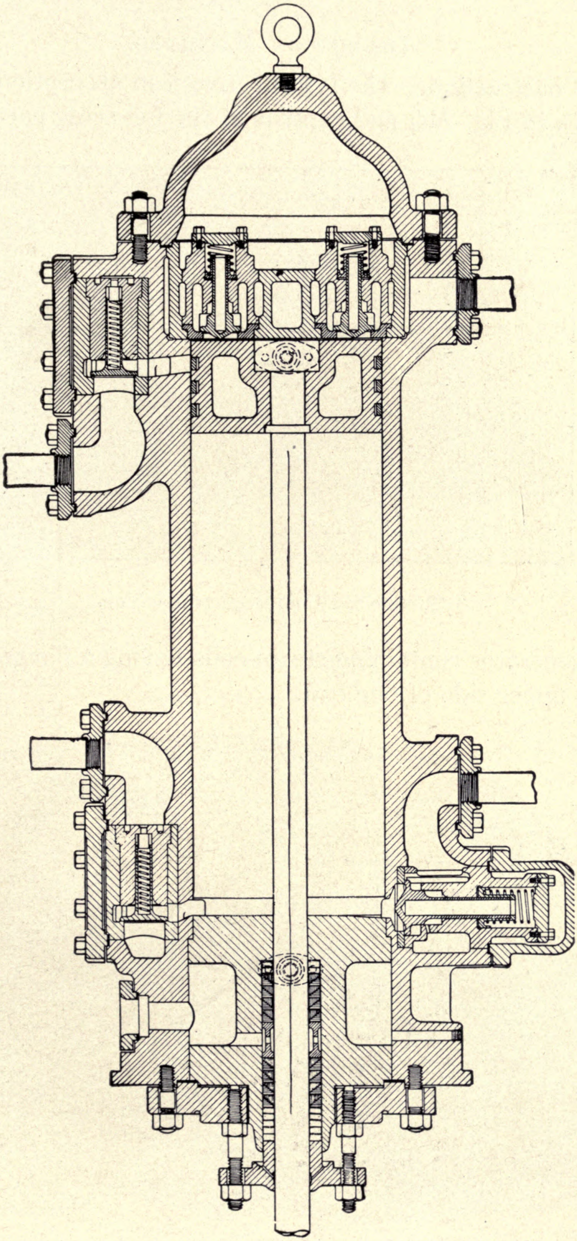
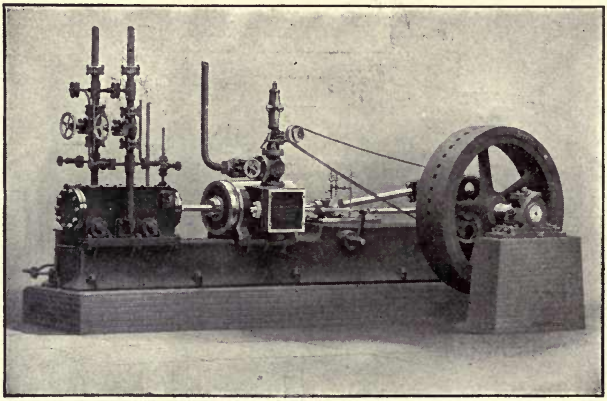


FIG. 243. **De La Vergne Machine Co.**

ABSORPTION MACHINES.

7. **Construction.**—The typical American absorption machine is shown in Fig. 244, and consists of the following parts:



Small-sized Compression Machine.

A generator containing steam-coils having a flanged opening on the upper side of one end.

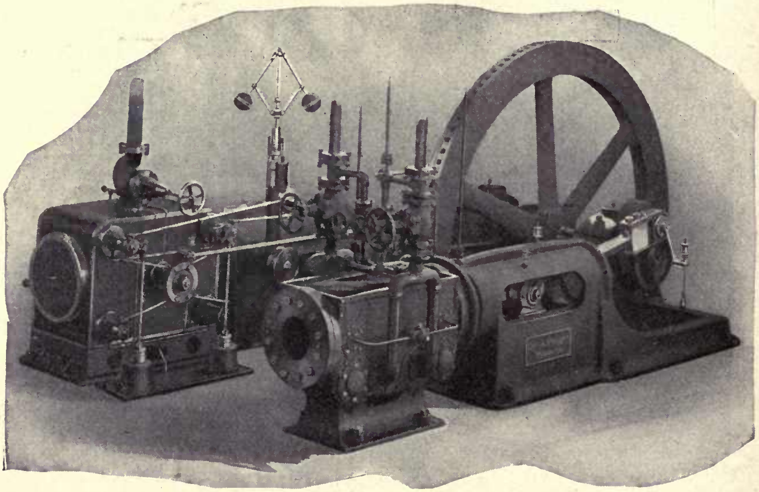
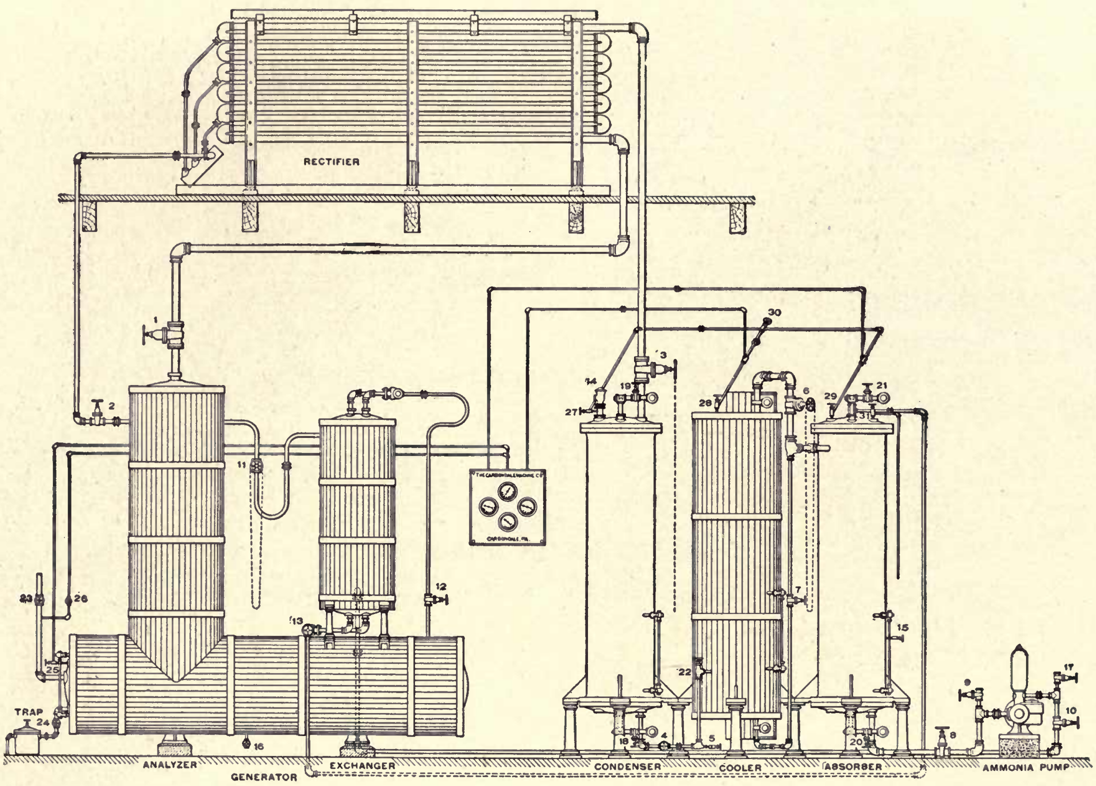


FIG. 245. DE LA VERGNE MACHINE CO.



Carbondale Machine Co.

FIG. 244.—Absorption Machine.

To face page 330

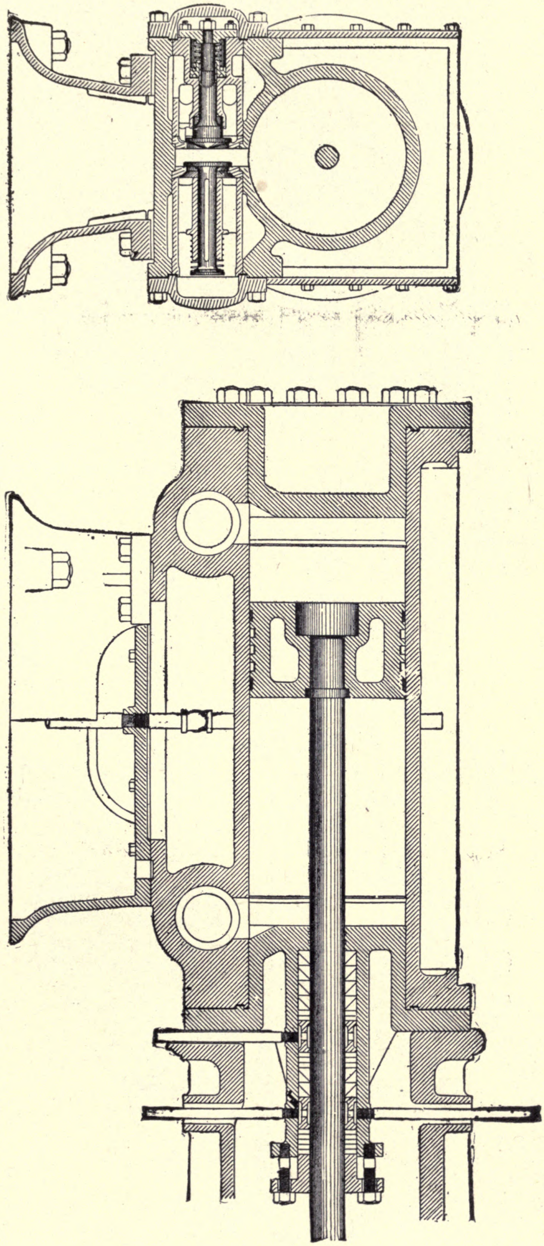


FIG. 246.

DE LA VERGNE MACHINE CO.

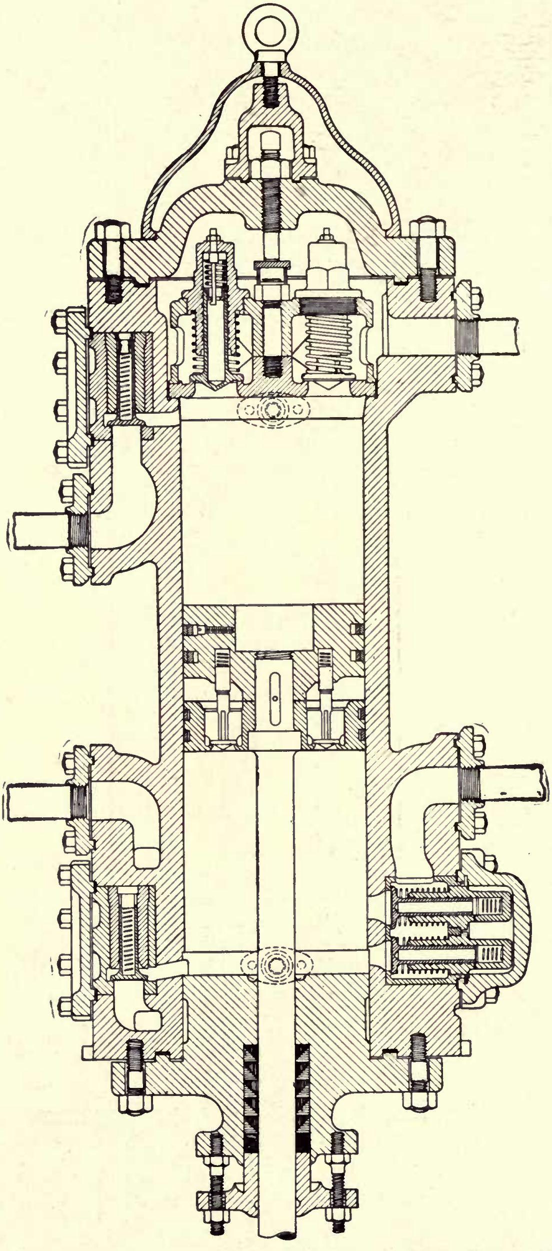


FIG. 247. **DE LA VERGNE MACHINE CO.**

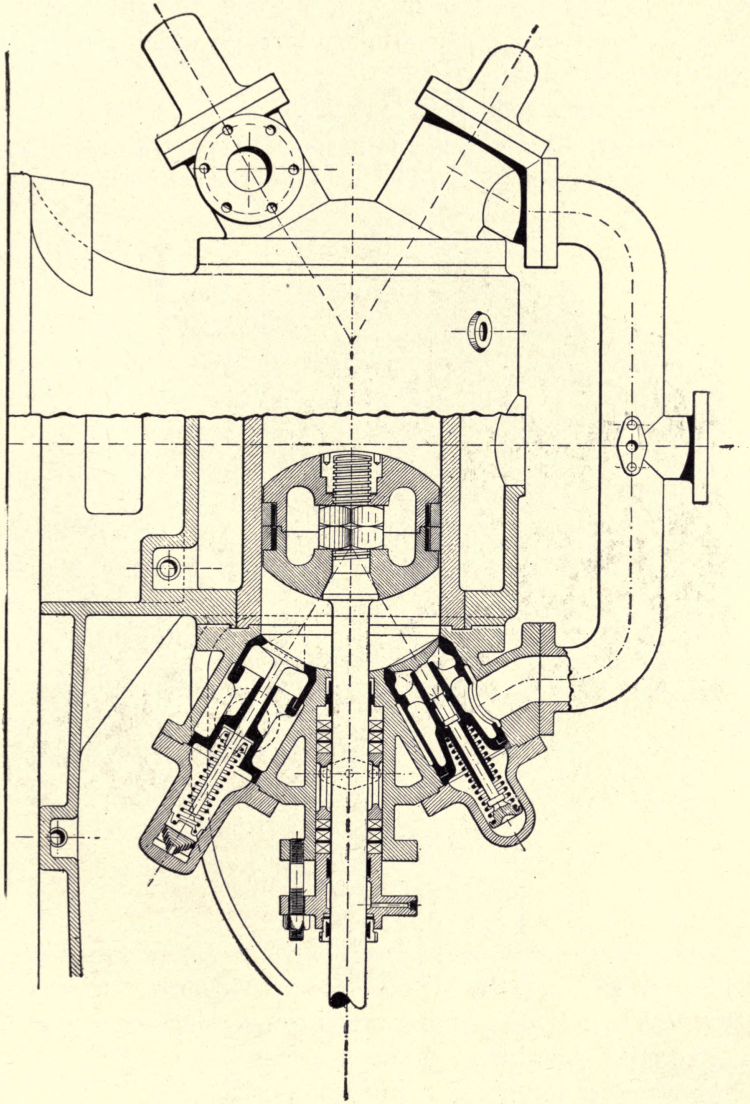


FIG. 248.

FRED W. WOLF CO.

An analyzer resting on this flanged opening and fitted with cast-iron trays with openings through which the gas from the generator rises and other openings through which the strong liquor from the absorber trickles down.

An exchanger containing spiral coils.

A rectifier, usually of the atmospheric type, with sprinkler-trough over each coil for the distribution of the condensing water.

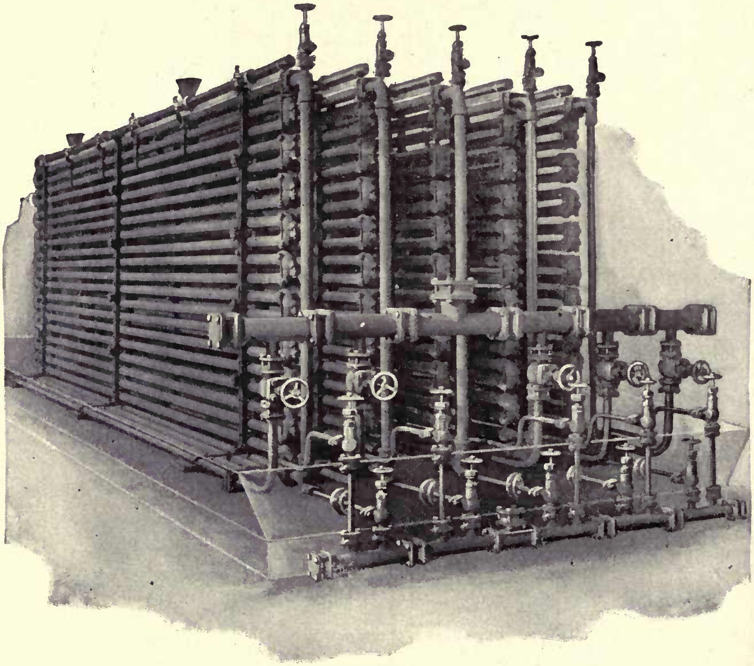


FIG. 249.

A condenser either of the atmospheric or the enclosed type, the former with a sprinkler-trough and distributing strips, the latter with spiral coils through which condensing water passes.

A receiver containing spiral water-coils.

An absorber containing spiral water-coils.

An ammonia-pump.

8. Operation.—The generator contains a solution of strong ammonia liquor in which the steam-coils are immersed. The ammonia in solution, having a lower boiling-point than the water, is partially vaporised by the heat from the steam-coils, leaving a weak solution of ammnoia. The gas thus liberated passes through the analyser to the rectifier. Whatever water-vapour may have been carried along with the ammonia-gas is condensed here and drips back into the generator.

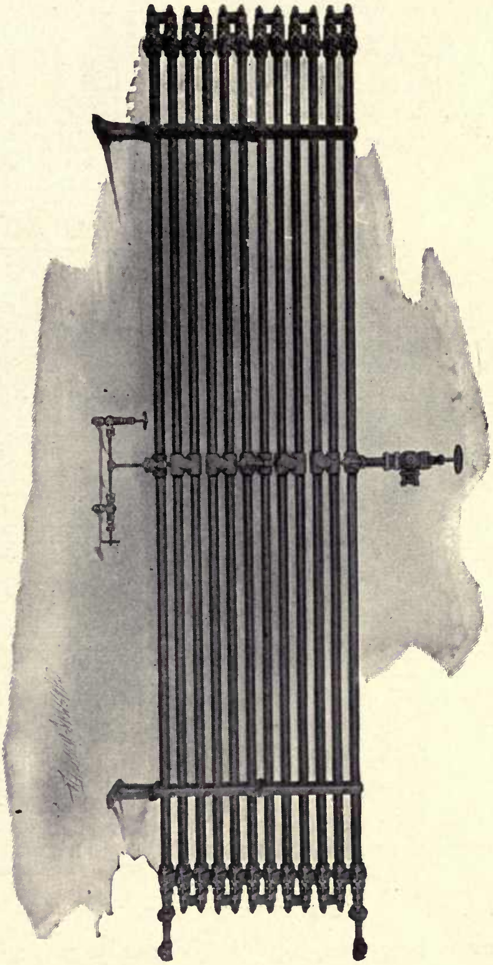
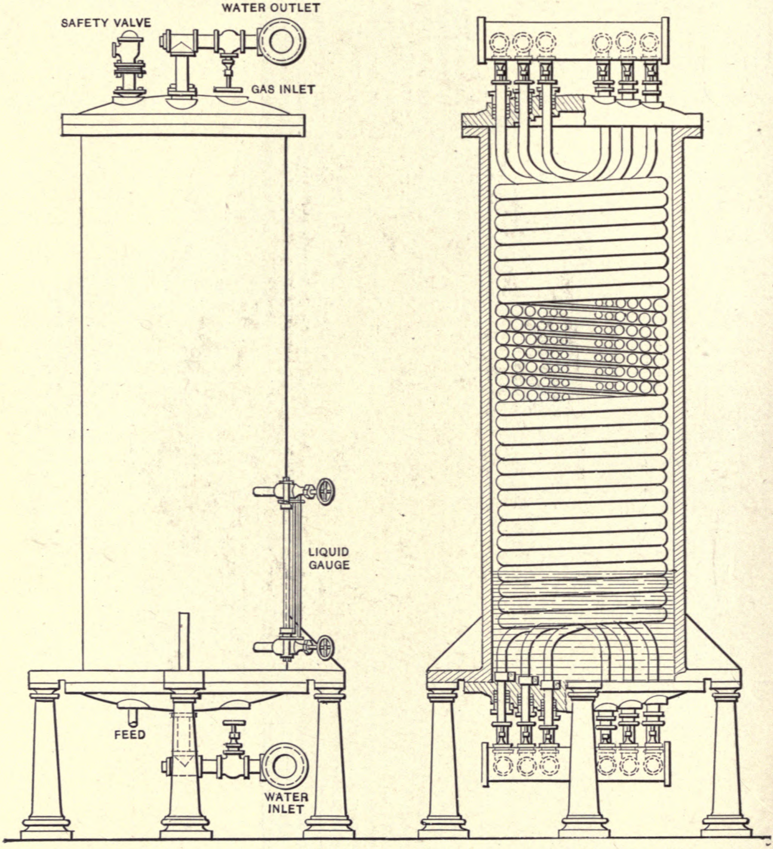


Fig. 250.

From the rectifier coils the gas passes into the condenser, is condensed, drains, and is collected in the receiver, from which it is expanded into the cooler or refrigerator coils. The gas from the cooler passes to the absorber and there meets the incoming weak liquor from the generator and is absorbed, forming strong liquor. This strong liquor is pumped through the exchanger into the top of the analyser and runs down over its pans to the generator.



Carbondale Machine Company.

Fig. 251.

It is desirable to have the strong liquor reach the generator as hot, and the weak liquor reach the absorber as cool, as possible. The exchanger is interposed between the generator and absorber in order that the weak and strong liquors may interchange their heat.

The pumps used may be of the direct-acting or gear-driven types.

9. **Ammonia Condensers.**—There are several types of ammonia condensers used in American practice—the atmospheric, the concentric or double pipe, the enclosed, and the submerged.

The atmospheric condenser, Fig. 249, of the "York" pattern consists of pipe coils which contain the ammonia and over which the condensing water flows; a pan or water-proof floor underneath collects and carries the water to waste. The condenser is often placed on the roof of the power-house or other exposed location in air-currents, which increase the evaporation of the condensing water and lower the temperature of the condensing surface.

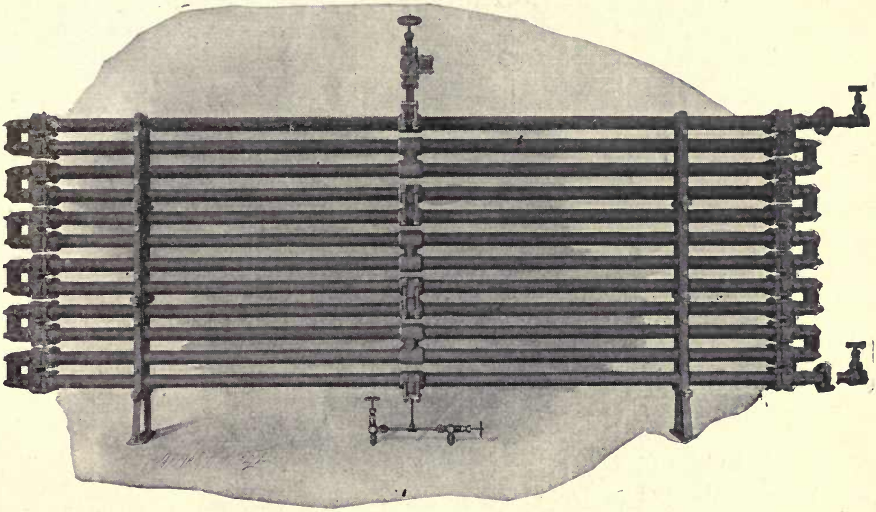


Fig. 252.

The "York" concentric or double pipe condenser, Fig. 250, consists of two pipes, one within the other. The inner pipe contains the condensing water, and the ring between the two contains the ammonia.

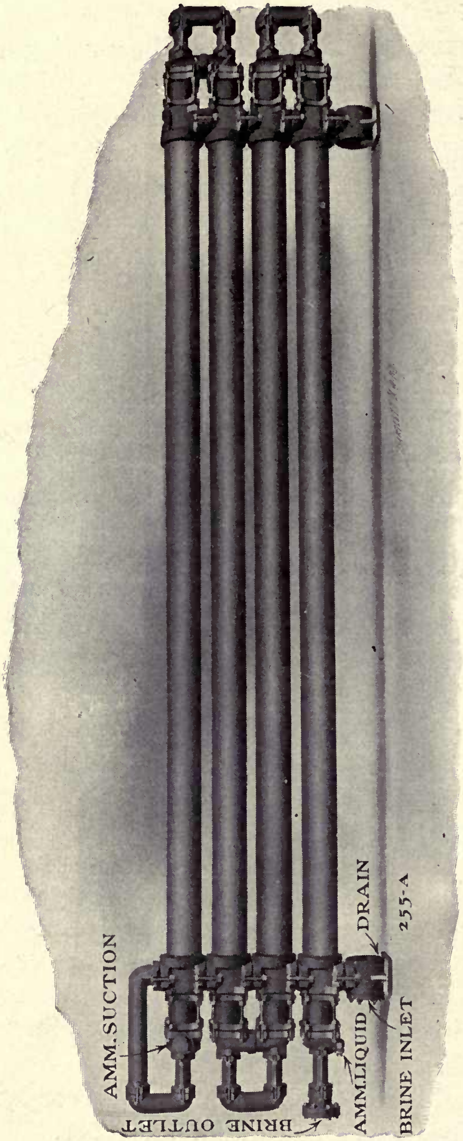


FIG. 253.

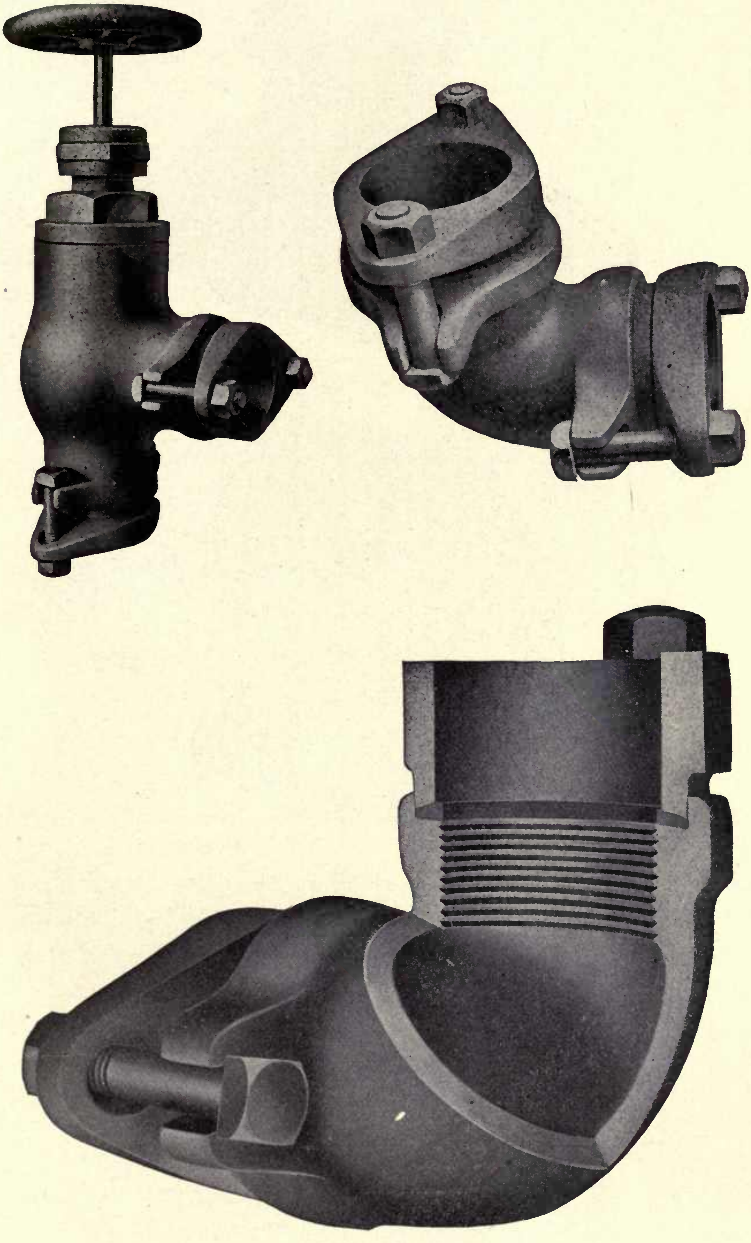


Fig. 254

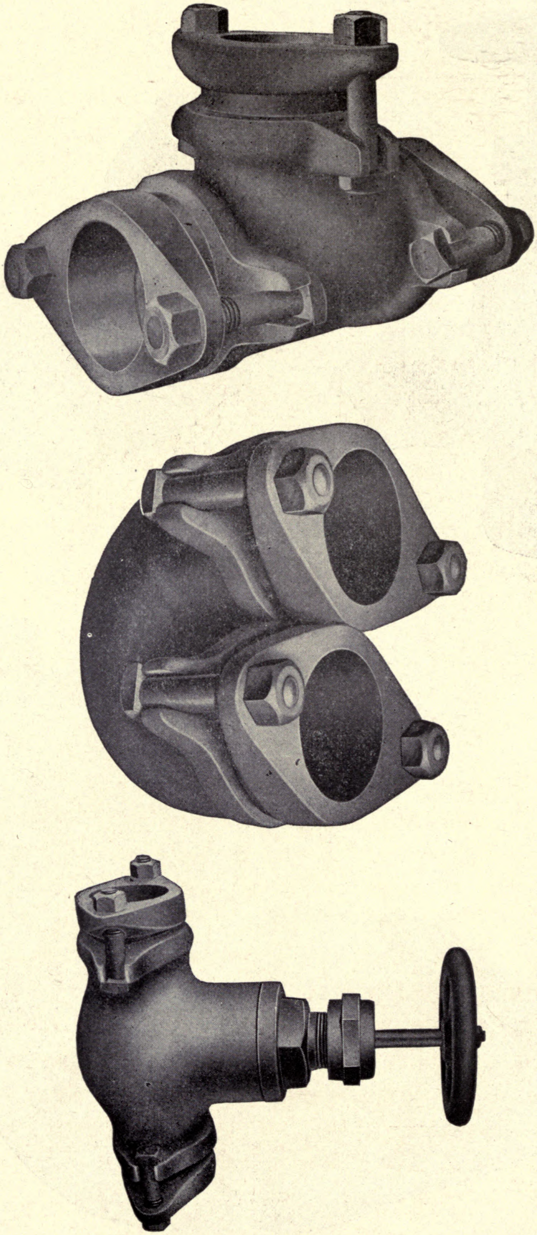


Fig. 254*a*.

In operation the water and ammonia flow in opposite directions, the cold condensing water being brought next to the liquefied ammonia. All condensing water is contained by piping, thus doing away with the necessity of using a pan or water-proof floor and allowing its use in any convenient location. While it has a large number of joints, the surface required is claimed to be less than for the atmospheric type.

The enclosed condenser of the Hendricks type, as in Fig. 251, consists of a cast-iron shell containing cooling-coils, the condensing water passing through these and the ammonia occupying the space between the shell and the coils; the coldest condensing water enters the coils at the bottom and is therefore in contact with the condensed ammonia. This type has the advantage of having all of the water contained by piping of few joints and occupies but little floor-space.

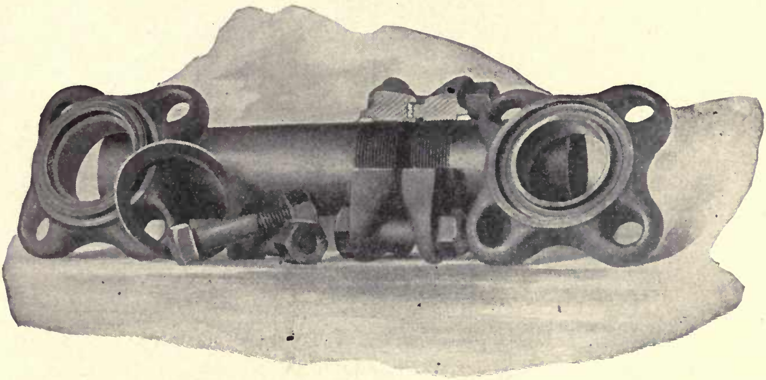


FIG. 255.

The submerged condenser consists of a series of coils containing the ammonia, the coils being submerged in flowing water. This has been largely superseded by the preceding types.

10. Brine-coolers.—Brine-coolers have gradually supplanted brine-tanks and are of two types, viz., the shell pattern with coils and the concentric tube type, being similar in design to the shell and concentric tube condensers.

The Hendricks brine-cooler, similar to the condenser shown in Fig. 251, consists of a cast-iron or steel shell containing spiral coils which contain brine, the ammonia being expanded into the space between the shell and the coils. The gas leaves the shell at the side near the top.

Shell coolers with straight tubes similar to vertical boilers have also been successfully used. The "York" concentric tube brine-cooler with double pipes, as in Fig. 252, has an inner pipe containing the brine and a space between the inner and outer pipe for ammonia evaporation. The ammonia enters the cooler at the middle of the bottom coil, going each way in the annular space and thence to the next coil by a connecting fitting at the middle, and similarly up through the whole section to the gas header at the top. The brine travels opposite in direction to the ammonia.

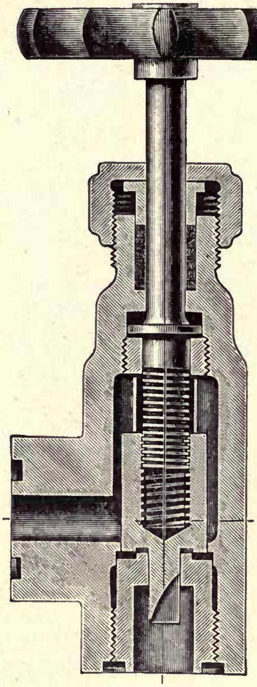


FIG. 256.

In the " Frick " type, Fig. 253, there are three concentric pipes, the inner and outer ones for brine and the space between the inner and second pipe for ammonia.

11. Pipe Joints, Flanges, Valves, and Accessories.—All piping intended to contain ammonia-gas or liquid should be carefully designed. Ammonia leaks are not only troublesome but are expensive, and any reasonable additional first cost may be justified by subsequent smaller operating expense.

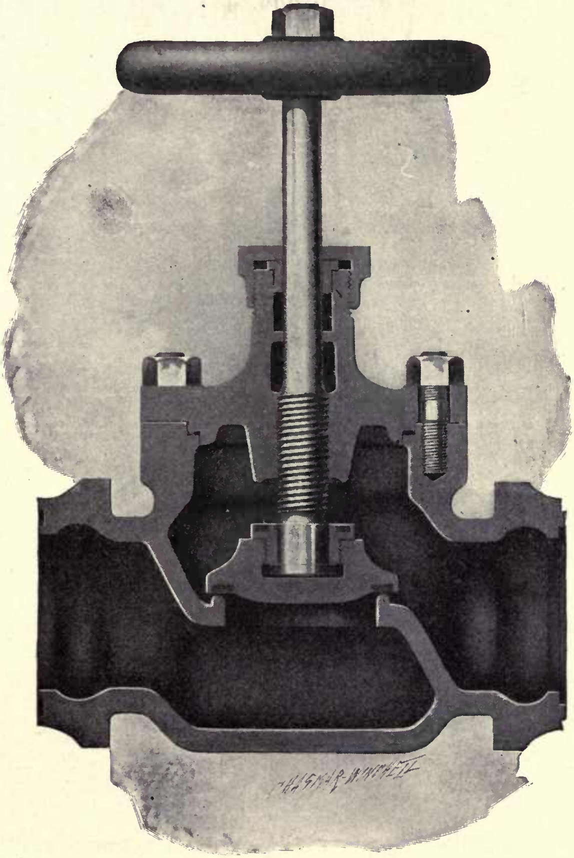


Fig. 257.

There are practically only two suitable types of pipe joints for ammonia, the gland and the De La Vergne types.

The former, similar to that shown in Figs. 254 and 254*a*, has a gland with packing in addition to the screwed joint. This packing can be renewed at any time and a leak by the thread easily stopped by tightening up the gland.

In the De La Vergne type, Fig. 255, in addition to the screwed joint there is an annular recess in the flange next the pipe. This recess and the pipe are tinned and the space filled with solder, reinforcing the thread and insuring a tight joint.

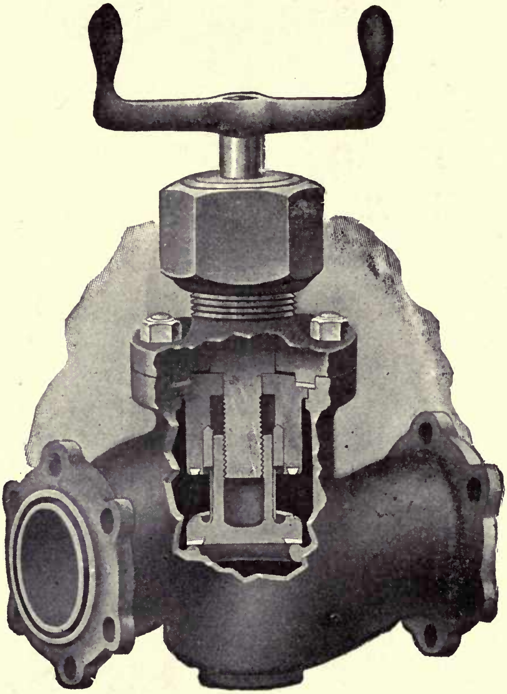


FIG. 258.

All flanges should be tongued and grooved, as in Fig. 255, with lead gaskets where the expansion is not excessive, and with a good rubber gasket where the expansion is likely to be considerable.

Automatic self-closing gauge-cocks are placed on all ammonia-receivers and other apparatus where it is essential to know the height of ammonia at any time.

All valves and fittings are made of mild steel or drop forgings.

Expansion-valves are of special construction, as in the "De La Vergne" type, Fig. 256. Other ammonia-valves may be either gate or globe, as in the "York" type, Fig. 257, and the "Triumph" type, Fig. 258.

Relief-valves are placed on any apparatus where excessive pressure may occur.

A certain amount of lubrication is necessary in all types of compression machines, and much more than is needed is used in nearly all plants. The separation of the oil from the ammonia-vapour before it reaches the condenser is of the greatest importance, as otherwise it is deposited there, cutting down its efficiency. It soon reaches the refrigerator or cooler with deleterious effect. Oil-separators similar in design to steam-separators with baffle-plates are used to intercept the oil.

CHAPTER VII.

AIR-COOLING.

1. **General Remarks.**—The necessity for keeping perishable goods in refrigerated compartments and the demand for cool atmospheres in auditoriums, dwelling-houses, factories, and other places has resulted in the adoption of several methods for air-cooling.

These are the induced or gravity, the forced or indirect circulating, and a combination of both.

It has been found that gas emanates from perishable products whenever they are stored in rooms and is absorbed by the moisture in the air.

The atmosphere contains mould or fungus germs which multiply rapidly in the presence of water-vapour, and it is essential, therefore, that there shall be a circulation of air within the compartment, carrying the air to the coils, in order that any vapour contained in it may be frozen out, drying and at the same time purifying the air by the removal of the vapour and with it the absorbed gases.

2. **Induced Circulation** may be accomplished by properly piping the compartment.

The best methods of direct piping for induced or gravity circulation are as shown in Figs. 156 and 158 of the first section of this book.

Slight modifications of these methods are widely used throughout the Uinted States.

3. **The Indirect Circulation** is by a fan system which passes air over coils in a room external to the compartment to be

cooled, supplying the air to, and exhausting it from, the refrigerator.

Both systems are used in refrigerated warehouses. For fruit and egg storage nearly all large modern installations have indirect air circulation similar to that described under "General Methods of Refrigeration."

The function of such a system is to cool, dry, and satisfactorily distribute the cold air to the compartment so that a uniform temperature and a satisfactory humidity may be maintained. The coil-rooms for such a system consist, as in Fig. 259, of a circulating fan, refrigerating-coils, drying-racks for fused chloride of calcium, and proper inlet and outlet flues for the air.

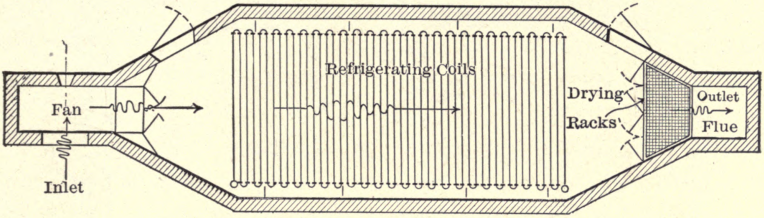


Fig. 259.

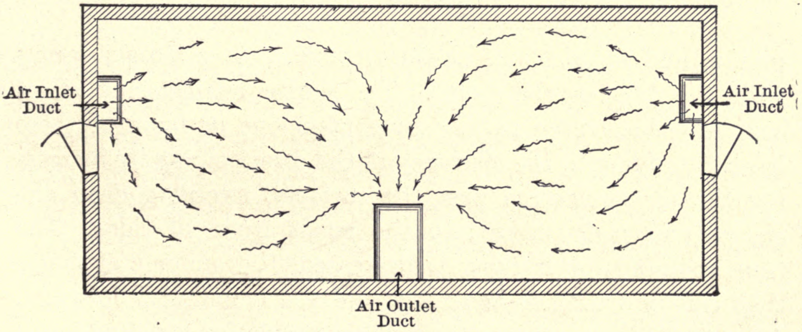


Fig. 260.

In distributing air to refrigerated rooms in warehouses two methods are extensively used—one, in which a large volume of air is admitted, usually at two points, in a room, sweeping it toward an outlet intermediate between the inlets, as in Fig. 260, or as in Fig. 261, by introducing the air through carefully designed flues, distributing it through numerous small openings and exhausting it at numerous other points or at some one point, as at *B*, at the end of the room and near the floor. Sharp freezers are often similarly operated.

4. Combined Induced and Indirect Air Circulation.—In the sharp freezing of meats and similar products, systems which combine direct piping with indirect air circulation are being introduced. The freezers are piped in the usual way for direct piping, and in addition, air-flues connect these storage-rooms with the bunker coil-rooms. The entire refrigerating effect can be concentrated at any time upon one or more of these freezers, adding to the efficiency of the direct piping by increasing the air circulation past the surfaces, sweeping dead air-spaces and penetrating into the goods from all sides. The time required for freezing is decreased, the amount of goods that can be frozen is increased, and the handling is lessened. After the goods are frozen the indirect circulation is discontinued and the room used as a holding freezer.

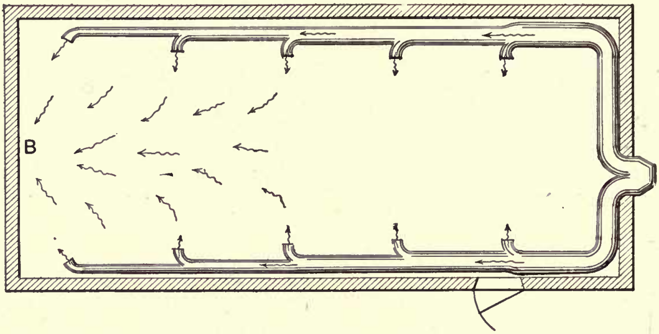


FIG. 261.

5. Coil-room Design.—The general plan of a coil-room having brine bunker coils for the indirect circulation system is as shown in Fig. 259.

The fans used for air circulation are overhung with bearings out of contact with the air, eliminating all contamination by oil.

The bunker rooms have water-proof floors usually constructed of 2-in. plank covered with No. 16 galvanized sheet iron with joints soldered and floors drained to a catch-basin; the walls are usually covered with No. 26 galvanized iron, lapped, with red-leaded joints; the ceiling is of sheet zinc with lock joints red-leaded. No surfaces exposed to air-currents are painted.

For the best efficiency, the piping consists of horizontal coils made up of 2-in. pipe with return bends 6 ins. on centres, extending from the fan to the outlet flue, as shown in Fig. 259, the vertical distance from coil to coil being reduced to 4 ins. or thereabouts if necessary.

All coils are staggered to obtain more effective cooling surface and are made up of pipe varying in length from 10 ft. to 20 ft.; all ironwork, such as pipes and pipe supports which come in contact with the air should be galvanized.

Coils varying in size from 300 to 500 lineal feet of 2-in. pipe are connected into supply and return headers, the brine flowing in a direction opposite to the air. With an air velocity over the coils of 1000 ft. per minute, the efficiency of the pipe surfaces may be taken as 4·0 B.T.U. per square foot per hour per 1° F. difference between the air and brine temperature.

Owing to the accumulation of frost on the brine-coils auxiliary heating-pipes are installed. With the comparatively short coils described, it is easy to maintain an even temperature in the storage-rooms by shutting off or turning on some of these coils as the work may demand.

In practice, especially after the goods are once cooled, only a small proportion of the coils are used, and when some of these are heavily frosted, they are turned off and the clean coils added to replace them; if all have thus been used, it is time to defrost by the auxiliary heating-coils.

In determining the amount of piping required, we can use the rules given under " Piping and Refrigeration Required," or upon a British thermal unit basis as described under "Auditorium and Chocolate Factory Cooling."

6. Sheet System.—In abattoir work the indirect method of air circulation, also direct piping with overhead bunker coils, is used for chill-rooms. In some installations, however, the sheet or curtain method has been used. In this method curtains or sheets of cheese-cloth, 8′×10′, over which chilled brine solution flows from troughs above, are suspended vertically, one in front of the other, in a large room similar to the bunker-room. The air in passing these curtains is chilled and dried. This system, however, is not ordinarily installed in storage-warehouses.

7. Linde System.—In the Linde system the bunker coils have a solution of calcium-chloride brine flowing continuously over them, thus preventing frost from accumulating and at the same time drying the air by the absorption of moisture.

8. Auditorium Cooling.—In cooling which relates to the refrigeration of material in storage-warehouses, where rooms have no regular occupants, the closed system is generally used. The air passes around successively from coil to storage-room with only occasional renewal. The refrigeration required after the rooms have been once cooled, if the insulation is satisfactory, is small.

In cooling auditoriums and similar places where the rooms have human occupants, instead of using the closed system with occasional renewal, new air is constantly being taken in at the temperature and humidity of the outside atmosphere, cooled, dried, and distributed to the apartments. When the air is cooled its capacity for holding moisture decreases until the dew-point is reached, at which point it is saturated. Further cooling results in the precipitation of moisture. If all the outside air is cooled from the inlet to the desired temperature, the relative humidity will be too high for the health and comfort of the occupants of the rooms. Air may be dried by cooling it sufficiently to precipitate some of the moisture and afterwards raising its temperature again. In practice the proper temperature and humidity are obtained by passing part of the air over the cooling-coils and by-passing the remainder, mixing both later before delivering to the rooms. In some cases, however, where the rooms are large and high, the chilled air is forced into them through numerous small openings near the ceiling where the mixing of the warm and chilled air takes place. From the top it descends and replaces the lower and warmer strata of impure air, which leaves the room at the bottom.

9. **Refrigeration Required for Auditorium Cooling.**—An accurate determination of the refrigeration required for cooling auditoriums and rooms is extremely difficult owing to the lack of experimental data. We can, however, make approximate determinations for this sort of work which are of valuable assistance in design.

The essential results to be obtained are the maintenance of a satisfactory purity, temperature, and humidity for the air.

The purity may be maintained by changing the air sufficiently often, care being taken to introduce it into the room in such a manner as to prevent a sensation of chilliness on the part of the occupants.

In practice, 30 or more cu. ft. of air per minute, or 1800 cu. ft. per hour, is introduced for each occupant of the room and a sufficient additional quantity to care for the products of combustion from gas-jets and other lights.

For ordinary temperatures, one British thermal unit will raise 55 cu. ft. of air one degree.

An adult person at rest will give off 400 B.T.U. per hour.

For gaslights, German engineers assume that 675 B.T.U. per hour are given off for every cubic foot of gas used.

For these and other lights, the heat evolved per candle-power as used by the same authorities is:

Gas, ordinary burner	300	B.T.U.
“ Wellsbach	31	“
Electric incandescent	14	“
“ arc	4·3	“

In estimating the losses through exposed wall and window surfaces, we can use 0·25 B.T.U. for walls and 1·00 B.T.U. for glass surfaces per square foot per hour per one degree difference between the outside and inside atmosphere. These latter constants are considered by Prof. Carpenter as fair, being based on experiments by Wolf and Peclèt.

With these data at hand, we can determine (*a*) the heat evolved per hour by the people; (*b*) the heat given off by the gas, electric, or other lights; (*c*) the amount transmitted through wall and window surfaces.

Sufficient refrigeration must be supplied by the incoming air to asborb this heat and after mixing with the warm atmosphere of the auditorium produce a temperature of 65°–68° and a relative humidity of 70 per cent, or a condition as near that of a June atmosphere as possible.

In large rooms the cooled air is often admitted at 40° F. or lower, and in such instances the inlet openings should be numerous and of sufficient size to insure slow moving air-currents so deflected as to prevent sensible draughtiness. A velocity of 100 ft. per minute through the inlet openings is allowable for very high rooms, but this must be reduced to 30 ft. per minute or less if they are located near the occupants of the rooms.

Unless it is desirable to dry the air by the incidental warming which will take place in passing through the supply air-ducts, these should be insulated.

When the temperature of the air and the quantity which must be supplied to the auditorium has been determined, the refrigeration required can be estimated as follows:

Assuming a temperature of 90° F. for the outside air with a relative humidity of 80 per cent, a temperature of 30° for air in the bunker-room with 15° brine in the bunker coils, the work imposed upon these coils may be divided into two parts:

(*a*) To cool the air from 90° to 30°.

(*b*) To cool the moisture in the air from 90° to 32° and freeze and cool the same from 32° to 15°.

For example, if we circulate 100,000 cu. ft. of air per hour, the total heat absorbed by the coils in cooling it from 90° F. to 30° F. will be:

$$100{,}000 \text{ cu. ft.} \times (90° - 30°) \times \cdot 019 \text{ B.T.U.} = 114{,}000 \text{ B.T.U.}$$

One cubic foot of air at 90° F. and 80 per cent relative humidity contains 11·83 grains of water.

The air taken in at 90° F. when lowered to 30° F. is cooled below the dew-point, and when it leaves the coil-room is saturated. Air at 30° F. saturated contains 1·93 grains per cubic foot. Therefore the amount of moisture which will be deposited on the bunker-room coils will be

$$100{,}000 \times (11\cdot 83 - 1\cdot 93) = 990{,}000 \text{ grains,}$$

or

$$990{,}000 \div 7000 = 141\cdot 4 \text{ lbs.}$$

To cool one pound of moisture from 90° to 32° and freeze it upon the coils will require:

(*a*)	To cool the moisture from 90° to 32° = . . .	58·0	B.T.U.
(*b*)	" absorb the latent heat.	965·7	"
(*c*)	" freeze the moisture.	142·0	"
(*d*)	" cool the ice from 32° to 15° = (17×0·5)	8·5	"
		1174·2	"

For the 141·4 lbs. the total B.T.U. required will be

$$141\cdot 4 \times 1174\cdot 2 = 166{,}032 \text{ B.T.U.}$$

By adding the B.T.U. required to cool the air we have

$$114{,}000 + 166{,}032 = 280{,}032 \text{ B.T.U.}$$

If we assume that 75 per cent of the refrigerating effect is effectively transmitted from the machine to the work the refrigerating capacity required will be

$$280{,}032 \times 1 \cdot 33 = 372{,}443 \text{ B.T.U.},$$

or

$$372{,}443 \div 11{,}833 \cdot 3 = 31 \cdot 5 \text{ tons of refrigeration in 24 hours.}$$

It is advisable in any installations of this kind to so proportion the apparatus that reserve units are always available. Its utility is tested during the hot and humid months of summer, and in order to be useful, its operation must be continuous.

10. Chocolate Factories.—The product of chocolate factories which carry the process from the raw bean to the finished product consists of the manufacture of:

(*a*) Chocolate in bulk, in large cakes which weigh 10 lbs.
(*b*) German sweet and plain chocolate in small cakes.
(*c*) Chocolate and cocoa in cans.
(*d*) Chocolate confectionery of various kinds.

By the use of refrigeration, chill-rooms, which in the best practice are under the indirect circulation of air, are maintained at 30° F. or thereabouts, and the chocolate, as in (*a*), after being run into moulds is placed on racks in these rooms. With good air circulation, it requires from 4 to 5 hours to cool the cakes weighing 10 lbs.; in some cases, however, the time has been reduced, and prior to the adoption of this practice, the manufacture of chocolate was suspended during the summer months.

German sweet and plain chocolate is similarly cooled, only a short time, however, being required.

The atmosphere in the dipping- and packing-rooms, where the sweet chocolate and confectionery are manufactured and packed for shipment, is kept at 68° F. and relative humidity as near 70 per cent as possible.

The cooling of the air in these rooms differs from auditorium cooling in that the system is a closed one, the air being forced by a fan through the coils to the workroom and again returned by the suction-ducts to the bunker-room, only occasional renewal being used.

The purity of the room atmosphere is maintained by the absorption of any existing impurities by the water-vapour in the air and the subsequent removal by freezing this water-vapour as frost on the coils. Some leakage will take place from the outside, and provision is made for the further addition of any quantity of outside air which may be needed by air-ducts connecting with the blower suction. Ordinarily this connection is only periodically used.

The refrigeration required for this class of work may be estimated as follows:

We can assume to illustrate that 100,000 cu. ft. of air is circulated per minute, that the temperature of the workroom is 68°, and that 200 persons are at work.

The work imposed upon the refrigerating-machine consists in taking care of:

(1) The heat given off by the people at work.

(2) The moisture given off by the people at work.

(3) The inflow of heat through exposed surfaces.

(4) The cooling and drying of any new air admitted from the outside atmosphere.

(5) The cooling of the chocolate and goods in the workroom.

If we assume that 450 B.T.U. per hour are given off by each person at work, for 200 people we will have for

$$(1)\ 200\times 450=90{,}000 \text{ B.T.U.}$$

If each person by respiration and evaporation from the body gives off 0·5 lb. of moisture per hour, and if we allow 1175·0 B.T.U. for every pound of moisture condensed, cooled, and frozen upon the coils, we have for

$$(2)\ 200\times 0{\cdot}5\times 1175{\cdot}0=117{,}500 \text{ B.T.U.}$$

If we assume there are 3000 sq. ft. of exposed wall surface and 100 sq. ft. of window surface with an outside temperature of 90° F. and a room temperature of 68° F., using 0·25 B.T.U. as the loss per square foot of wall surface and 1·0 B.T.U. as the loss per square foot of window surface per hour per 1° difference, we have for

$$(3)\ [(3000\times\cdot25)+(100\times1)][90-68]=18{,}700 \text{ B.T.U.}$$

If we assume that the specific heat of chocolate is ·90, and if 500 lbs. are cooled per hour through a 20° range, we have for

$$(5)\ 500\times\cdot9\times20=9000 \text{ B.T.U.}$$

The work to cool and dry new air can be estimated as described under "Auditorium Cooling."

Adding 1, 2, 3, and 5, we have as a total 235,200 B.T.U. per hour, or 20 tons, nearly, of refrigeration for 24 hours.

If we allow an efficiency of 75 per cent for the refrigerating apparatus, we will have 27 tons, approximately, as the refrigerating capacity required.

As in auditorium cooling, the lack of sufficient experimental data makes any estimate only an approximation to guide in design.

Ordinarily the temperature of the dipping-room is from 65° to 70° with a relative humidity of 70 per cent to 75 per cent.

The temperature of the inlet air from the bunker-room is usually about 50°.

CHAPTER VIII.

SPECIAL APPLICATION OF REFRIGERATION.

1. **Pipe-line Refrigeration.**—In several cities of the United States, compartments in well centralized market districts are refrigerated from central power plants. In such instances refrigeration has been successfully supplied by pipe-lines, often of great length, which carry the cooling medium from the central plant through the refrigerators and back again for recooling. This medium may be either ammonia or chloride-of-calcium brine. The largest and most successful installations are those in which the latter is used.

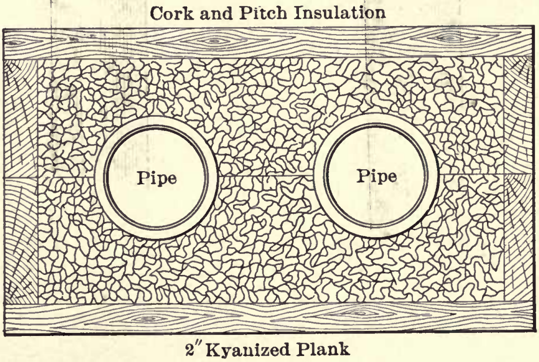


FIG. 262.

Owing to the frequent opening of doors in market compartments and the consequent inrush of warm air together with frequent exceptional exposures, the refrigeration required is more than that needed in warehouse practice. The economy and utility of installations of this character depend upon the

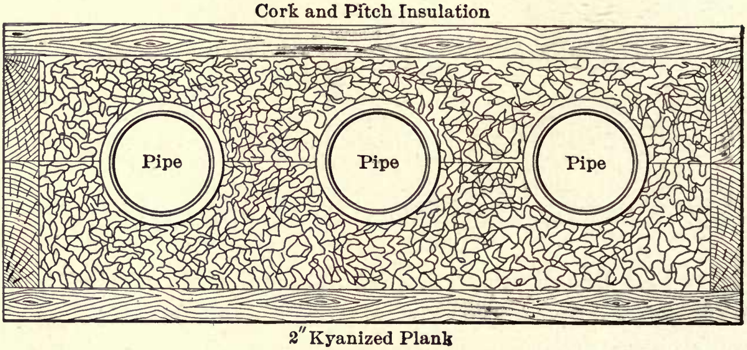


FIG. 263.

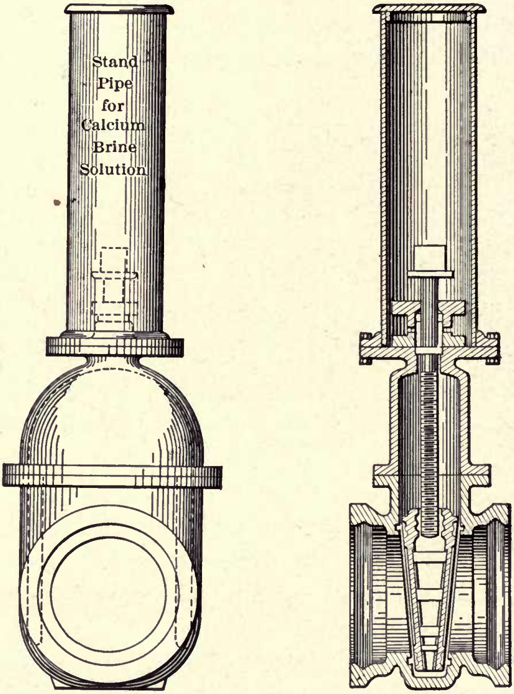


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FIG. 264.—Main Shut-off Valves.

equipment in the power-house, the proper piping in the street, and the insulation.

The supply and return mains are usually of cast iron, bell, and spigot pipe for sizes greater than 3 ins. diameter, insulated as in Fig. 262 and 263 and encased in boxes of kyanized plank. The smaller pipes are of wrought iron.

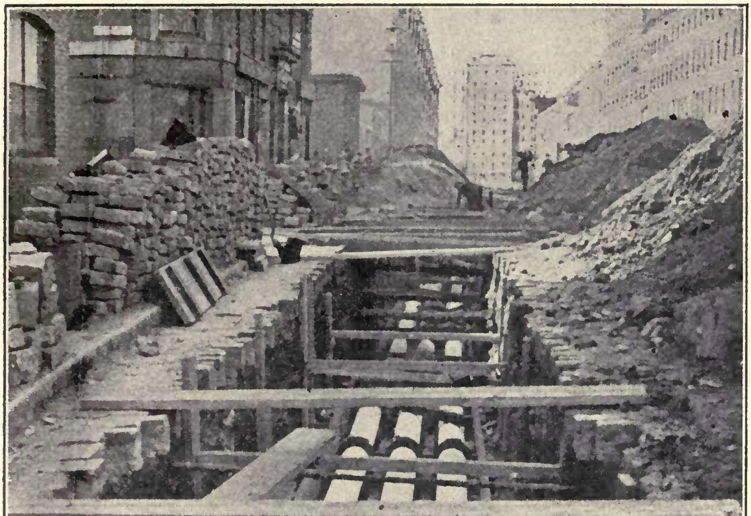
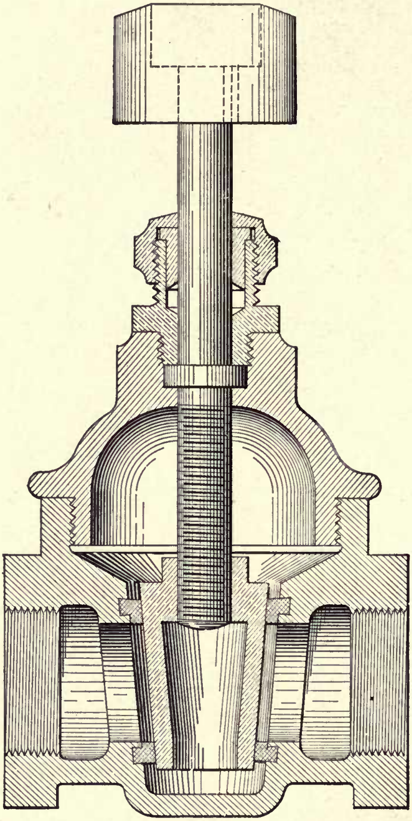


Fig. 265.—Three-pipe Brine System.

Three pipes are sometimes used—a supply, a return, and a spare or emergency pipe, as in Fig. 263 and Fig. 265.

Cast-iron pipes should have deep bells, with the joints made up first by driving in against the spigot end a lead pipe whose diameter equals the thickness of the calking space and of sufficient length to just encircle the pipe. This is followed by Selden packing tamped against the lead pipe, the latter preventing the packing from entering the pipe itself. The remaining space in the bell is calked with lead. A joint of this



Chapman Valve Mfg. Co.

Fig. 266.—Service Valve.

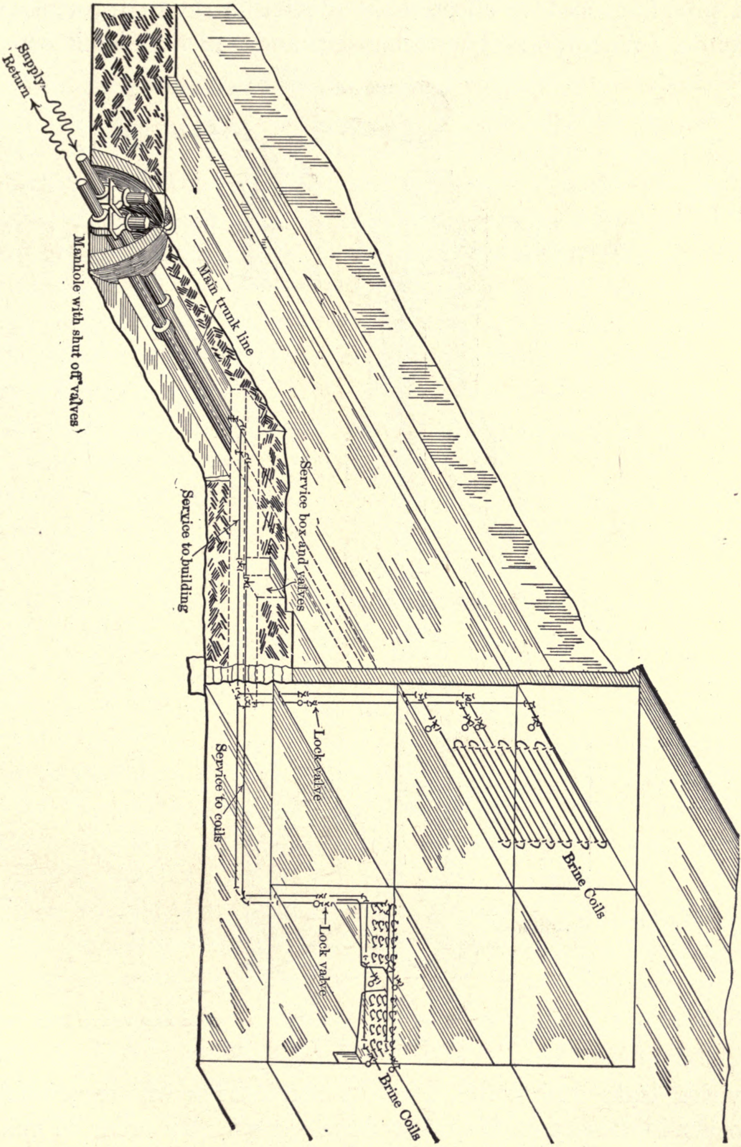
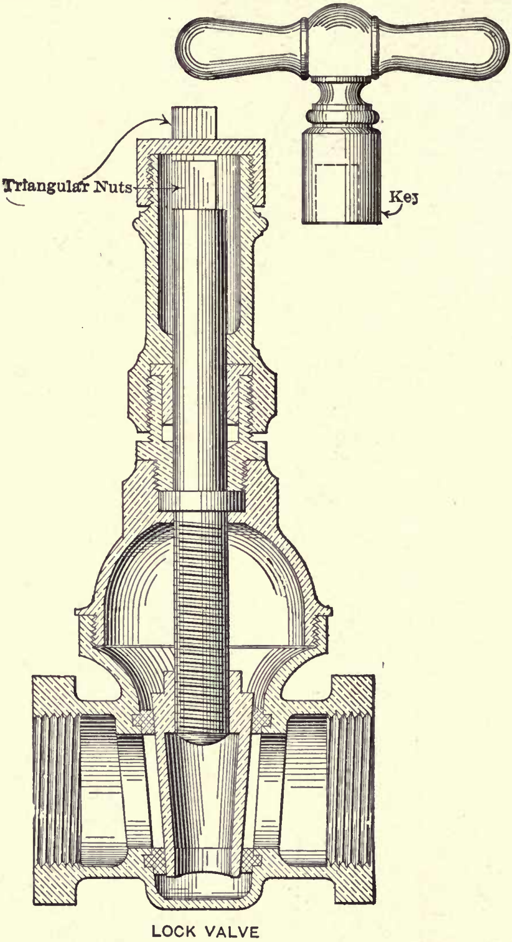


FIG. 267.—Pipe-line Brine Distribution.

description withstands the expansion and contraction in such a pipe-line, and is effective in preventing leaks through the joints. With the best workmanship and under constant supervision leaks due to fire, frost, or carelessness will occasionally occur in some part of an extensive system. In order to guard against large losses of brine and long interruption of service when leakage occurs, the trunk lines, as shown in Fig. 267, are



Chapman Valve Mfg. Co.

Fig. 268.—Lock-valve

divided into sections by gate-valves constructed as shown in Fig. 264, and contained in brick manholes. When leaks occur they can be traced to one section, isolated and repaired.

Owing to trouble from frost which accumulates about the head of the valve-stem and the consequent trouble in exigencies where quick closing is necessary, a stand-pipe, Fig. 264, containing a calcium brine solution surrounds the nuts on the valve-stem and keeps them clear for the closing wrench.

Branches of about 1½ ins. diameter of extra-heavy wrought-iron pipe extend, as in Fig. 267, from the top of the street mains to the service-valves, which are located just outside the building containing the refrigerator. These branches are insulated with pitch and cork and are encased in kyanized plank boxes. Service boxes extend from the valves to the street level and have cast-iron covers.

Gland fittings similar to the ammonia gland fittings, with extra-heavy wrought-iron pipe, are used for the services as far as the service-valves, which are constructed as in Fig. 266. Beyond these common pipe with long turn fittings is used. The service mains to the refrigerators in the buildings have gate-valves on the supplies and lock-valves, as in Fig. 268, on the returns, by which the quantity of brine to the compartments is regulated. Each valve is operated by inspectors employed by the corporation supplying the refrigeration. Generally the coils in the refrigerators are of 2-in. pipe made with return bends usually 6 ins. on centres, and with gate-valves both on the inlet and outlet.

There are plants in the United States in which the maximum tonnage required for such a system of brine distribution is 400 tons of ice-melting effect in 24 hours for summer service. Three million gallons of calcium-chloride brine with a specific gravity of 1225 and having a temperature of 14° F. at leaving and 18° F. upon returning to the power-house is distributed to several hundred boxes or compartments. The compartments vary in size from 200 to 12,000 cu. ft. capacity and are maintained at 35° F. or lower.

The choice of calcium-chloride brine as the refrigerant is due to the safety and ease with which it can be handled and to the absence of corrosive action on iron pipes. In large systems which have warehouses in addition to the pipe-line refrigeration, fused calcium chloride is often used for air-drying purposes. This calcium in absorbing moisture from the air becomes dissolved and is then used in the circulating system as needed.

General Design.—In designing a street system such as has just been described, the two principal elements which need to be considered are the piping and the power equipment.

Piping.—The general arrangement, the kind of pipe, and the type of valves and fittings has just been described.

The amount of piping required for the market compartments is given under " Refrigerated Compartments." For easy manipulation and the maintenance of a constant temperature, the piping in these had best be divided into two, or more short coils in preference to a single longer one. Fig. 158, the first section of this book, with slight modifications illustrates the best method of installing coils. In operation the coils are regulated by the valves on the return, and these are never open more than a fraction of a turn; it is better, however, to have ample connections in order that they may be opened periodically and the coils swept of any sediment which may accumulate.

In the operation of a brine system the chief interruption to circulation is from the accumulation of the air at the high points in the piping. Air-tanks to collect this air should be installed at these points and may be blown off periodically or constructed to discharge the air automatically.

If a vacuum is maintained on any part of the system, air will necessarily be drawn in whenever purge- or air-valves are opened. This should be guarded against. Pipe areas should be ample, with easy bends, in order to reduce the friction as much as possible.

While the supply-brine temperature in several distributing systems is 12° or thereabouts, there is a tendency to adopt 0° as the initial temperature, and, moreover, systems which will operate with 12° brine when first installed require a lower initial temperature when sediment has accumulated in the system and the insulation of the refrigerators has depreciated.

Power Equipment.—The power equipment consists of two elements—the refrigerating and the pumping machinery with the necessary auxiliaries.

There are two possible, distinct kinds of equipment, one consisting of the compression type of machine, which for best economy must have the ammonia compressor operated by a compound condensing engine using 17 lbs. of steam or less per hour per indicated horse-power, with a pumping-engine equally efficient and the auxiliaries of an economical type relative to steam consumption. This necessarily means high first cost. The other alternative is the installation of the absorption machine, so designed that the steam supplying the generator is taken from the exhaust of the engine operating the brine pumps and the auxiliary machinery.

The relative efficiency of the two systems is yet to be demonstrated; the first cost of the latter is considerably less.

Having decided upon the character of the units it is equally as important to decide upon their size. In New England the refrigeration required for the winter months is approximately 33 per cent of that required for the summer season.

The pumping capacity does not vary materially, since it is necessary to maintain a good circulation and suitable pressure in the system at all times.

The units should be in duplicate, with spare ones available in case of accident. It is advisable to so proportion these that one will do the work of the winter season easily, the remaining ones being out of commission for overhauling and repairs preparatory to the exacting conditions of the hot and humid weather of summer.

2. **Fish-freezing.**—In plants not especially equipped for fish-freezing, the fish are either spread upon galvanized iron

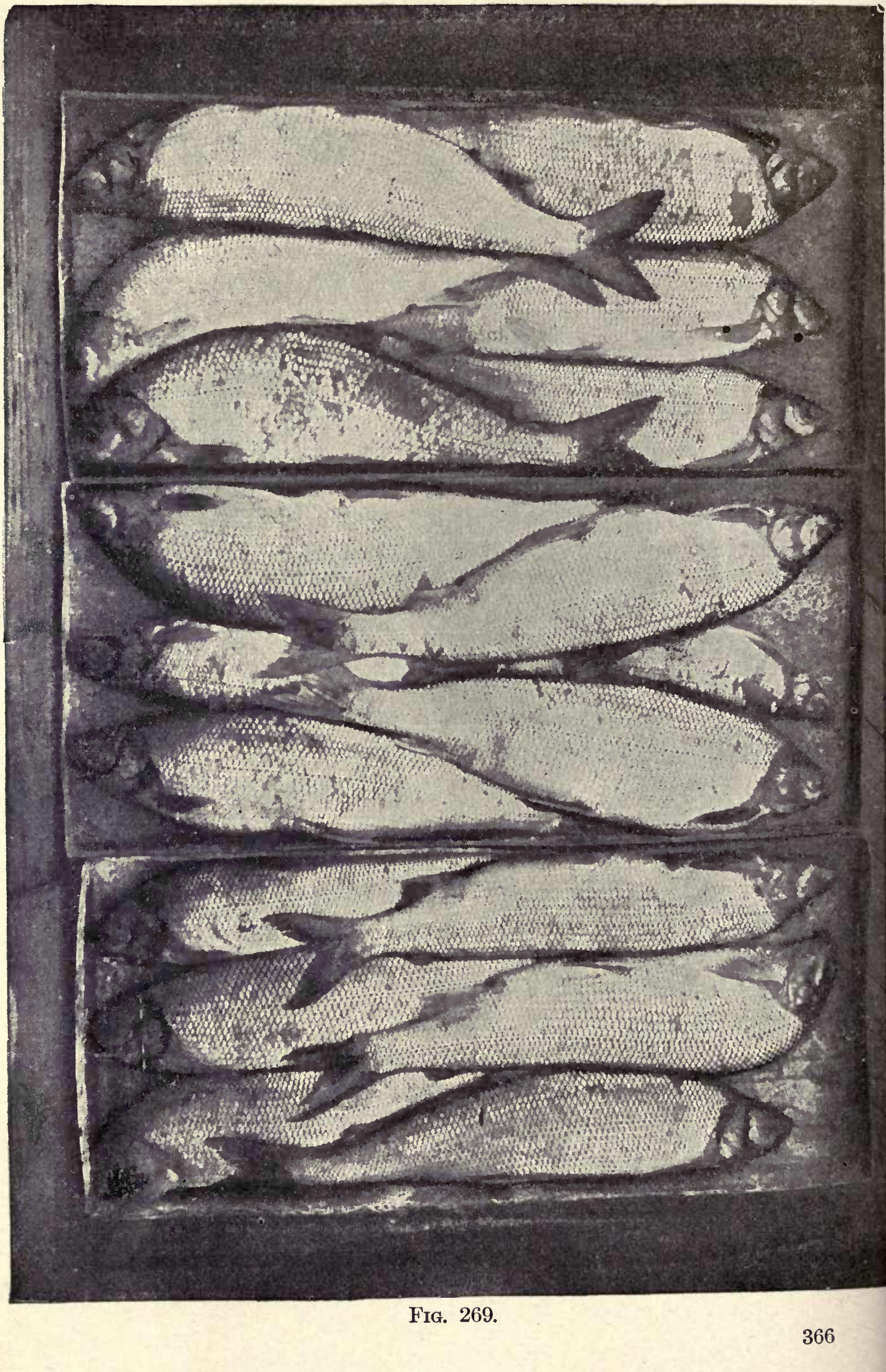


FIG. 269.

floors or hung upon hooks in the chill-room. This method is not only inefficient but requires an unnecessarily large amount of floor-space, labor, and time.

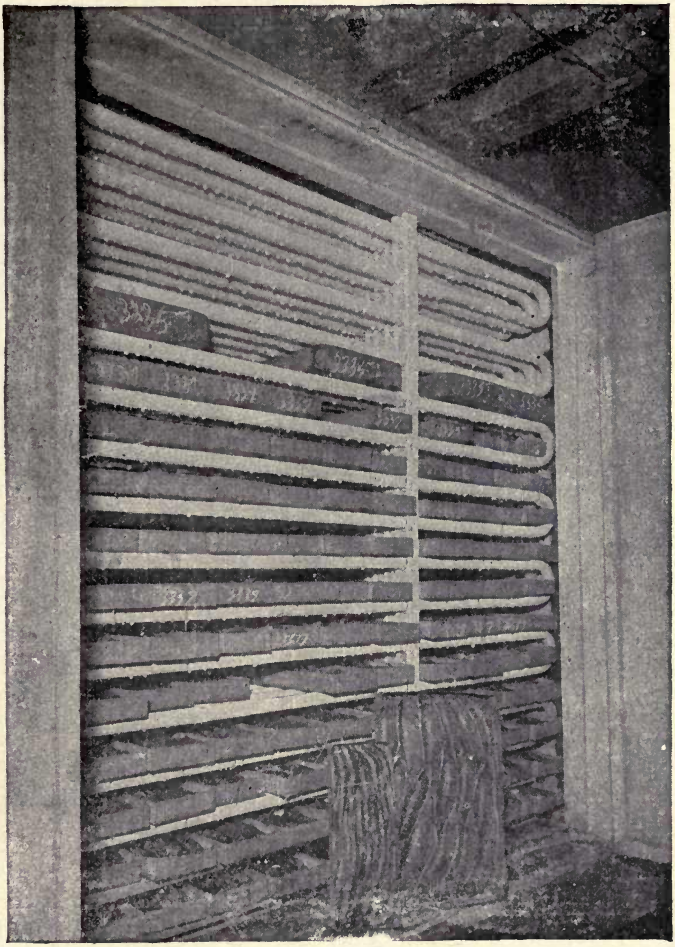


Fig. 270.

Better methods of freezing have been adopted in several parts of the United States where large quantities of fish are handled, the freezing being done in small sharp freezers heavily piped either with 1¼ in. diameter direct-expansion or with

2-in. diameter brine-pipe. The fish are placed in these freezers either on wooden trays or in galvanized iron pans, as in Fig. 269. The size of the pans varies with the size of the fish.

Freezers having a capacity of 5000 lbs. of fish have been equipped with 1500 ft. of 1¼-in. direct-expansion pipe made up in seven coils with pipe 6 ins. on centres, arranged 4 ins. on centres side by side, this arrangement forming shelves of pipe fifteen in number one above the other. The freezers are heavily insulated and enclosed by heavy doors in front, as in Fig. 270.

After being frozen and reduced to a temperature of 10° or more below zero, the fish are removed from the pans and passed through a bath of water, which glazes them with ice. They are then packed away in holding freezers. With a battery of ten such freezers 50,000 lbs. of fish can be frozen in 12 hours.

In salmon-freezing the fish are frozen separately, glazed with a thin coating of ice, and after being wrapped separately in paper are packed in boxes for shipment.

3. Water-cooling.—In hotels, apartment houses, offices, and other buildings it is becoming common io install refrigerating machinery to cool the drinking water. The system consists usually of a cooling-tank located in the basement near the engine-room. The water, which is cooled by submerged coils containing chloride-of-calcium brine, is circulated by a pump and passes up through risers to the drinking-fountains. These are connected in series through the building, and the risers terminate in two mains, one located in the basement and the other in the upper story. The unconsumed water in the upper main passes into a so-called balancing-tank which is somewhat higher than the main. From this the water overflows to the cooling-tank below, and the make-up supply is there furnished by a float-valve. The faucets should be as close to the risers as possible, and of course the risers can be made as circuitous as desired. The pump should have a capacity somewhat greater than the quantity of water used, and must be kept constantly in operation. The quantity of water allowed per person is about one gallon per day of 10 hours.

The water is supplied to the drinking-faucets at a temperature of 35°–40° F.

4. **Drying Air for Blast-furnaces.**[1]—Various methods of absorbing moisture from the air having proved insufficient, the Carnegie Steel Company recently installed a refrigerating plant at Etna, Pa., for the purpose of drying the air to one of its blast-furnaces. In effect, it consists of the system as described under "Indirect Air Circulation with Brine Bunker-Coils."

The air after passing the coils of the buker-room goes to the blowing-engines and thence to the blast-furnaces.

During a period of thirteen days the moisture in the atmosphere was 5·66 grains per cubic foot, and in the dried air 1·75 grains per cubic foot, while the amount of moisture removed per ton of iron produced was 69 lbs.

In operation, this plant with ordinary atmospheric air showed a production of 358 tons of iron with 2147 lbs. of coke per ton of iron, while with the dried air it showed 447 tons of iron with a consumption of 1726 lbs. of coke per ton of iron.

Before the use of refrigeration for drying the air, the horse-power for each blowing-engine was 900 I.H.P. After its use the horse-power was 671 I.H.P., or a difference of 229 I.H.P. per engine, with an aggregate saving of 687 I.H.P. for three engines.

The power required on the refrigerating end for the engines driving the compressors, blowers, brine and water pumps was 535 I.H.P., showing a net saving of 152 H.P. on the power end, together with an increased output at the furnaces with a smaller consumption of coke.

The installation consists of two 225-ton refrigerating-machines, one in reserve, 90,000 lineal feet of 2.inch brine-pipe, and other auxiliary machinery.

We append a table of one day's run showing the drying effects of the operation.

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[1] From a paper by Mr. James Gayley, before the Iron and Steel Institute in N. Y. City, and described in the Engineering Record of Nov. 5, 1904.

TEMPERATURE AND HUMIDITY IN THE REFRIGERATING-CHAMBER.

Time.	Temperature of the Air in Degrees F.		Grains of Water per Cubic Foot of Air.	
	Inlet.	Outlet.	Inlet.	Outlet.
6 A.M.	68	21	5·19	1·33
7 "	68	20	5·02	7·24
8 "	70	20	5·56	1·55
9 "	73	20	5·37	1·46
10 "	74	24	5·47	1·81
11 "	77	20	5·56	1·53
12 M	77	21	6·04	1·53
1 P.M.	80	21	6·04	1·42
2 "	81	22	6·14	1·60
3 "	81	23	5·74	1·60
4 "	82	23	5·74	1·55
5 "	82	22	6·04	1·62
6 "	81	23	5·94	1·55
7 "	80	23	5·74	1·62
8 "	79	24	5·94	1·55
9 "	73	23	7·01	1·85
10 "	73	22	6·78	1·70
11 "	73	23	6·78	1·70
12 Night	73	23	7·01	1·70
1 A.M.	73	23	6·78	1·70
2 "	74	23	7·01	1·70
3 "	73	23	6·78	1·70
4 "	73	23	6·78	1·48
5 "	73	23	6·78	1·48

5. **Marine Refrigeration.**—In marine work vessels in passenger and freight service are equipped with refrigerating machinery and chilled compartments maintained for meats and other perishable products. The holds of vessels used for fruit transportation are insulated and cooled by the indirect circulation of air, thereby keeping them at a constant temperature throughout the voyage. Excellent results have been obtained by this service and extensive equipment is under way for large fruit transportation companies. Steamers for the transportation of live cattle are equipped mostly with direct piping, in order to maintain the holds at a moderate temperature and prevent heavy loss of live stock.

6. **Hotel Refrigeration.**—The refrigerating-machine is looked upon as necessary in all large modern hotels for freezing and cooling purposes.

Compartments for wines and liquors, fruits, vegetables, poultry, game, butter, pastry, and miscellaneous articles are chilled. In addition, freezing-tanks for ice-making, compartments for freezing water in caraffes, freezing apparatus for ice-cream, and a water-cooling system for the drinking-faucets throughout the building are installed.

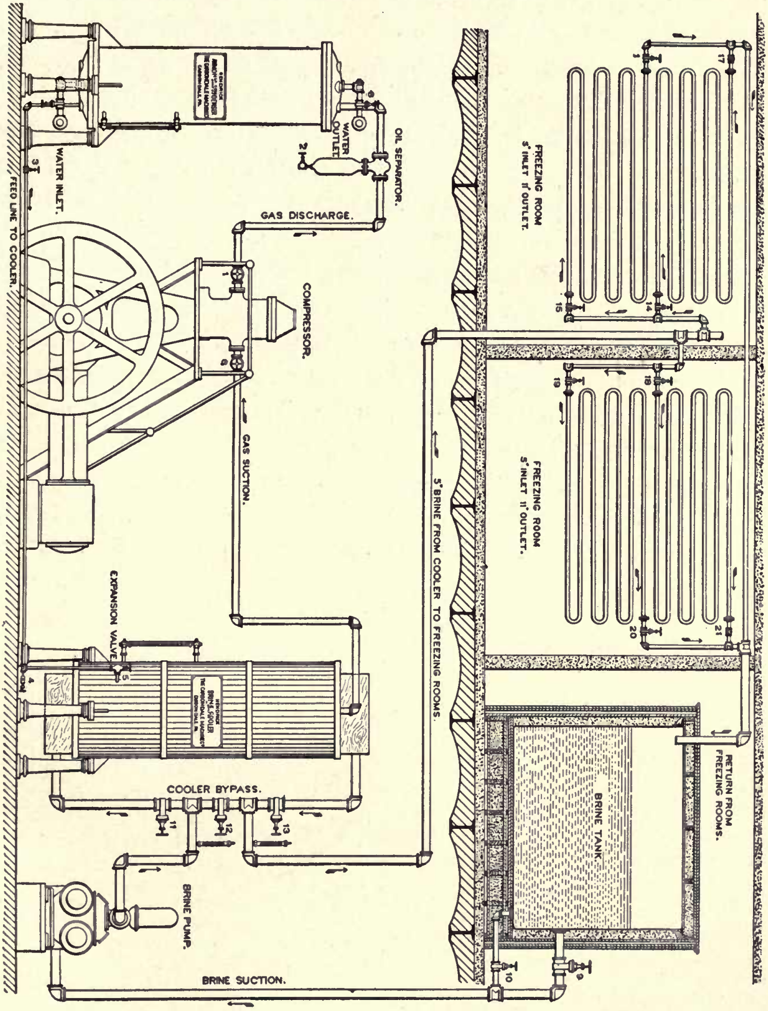

FIG. 271.

The availability of low-pressure steam exhausted from other apparatus, together with the necessity of machinery which shall be noiseless in operation, has led in some instances to the use of the absorption exhaust-steam system, using chloride-of-calcium brine as the chilled solution for circulation. In many cases compression machines are used as shown in Fig. 268, the brine-tank often being fitted with cans for freezing ice.

The ammonia condensing apparatus is necessarily the enclosed or double pipe type, and the condensing water after absorbing heat in the refrigerating-machine can be used for domestic service throughout the building.

7. **Other Applications.**—Refrigeration is also used extensively in candle and paraffin-oil works, butter factories and dairies, tea factories, dynamite works, in the manufacture of photographic plates, wine factories, soda-water works, sugar refineries, chemical works, india-rubber works, laundries and morgues, in tunnelling and sinking shafts, together with numerous other miscellaneous smaller applications. Among these may be mentioned the preservation of furs from the ravages of moths, the temperature for this purpose being maintanied at 28° F. and below.

APPENDIX.

TABLE A.

CALCIUM CHLORIDE.

The following table gives the properties of calcium chloride.

Degrees Baumé at 60° Fahr.	Specific Gravity at 60° Fahr.	Per Cent of Calcium.	Freezing-point, Degrees Fahr.	Degrees Baumé at 60° Fahr.	Specific Gravity at 60° Fahr.	Per Cent of Calcium.	Freezing-point, Degrees Fahr.
1·0	1·007	1	31·10	21·0	1·169	19	+ 1·76
2·0	1·015	2	31·38	22·0	1·179	20	− 1·48
3·0	1·024	3	29·48	23·0	1·189	21	− 4·90
4·0	1·032	4	28·58	24·0	1·199	22	− 8·68
5·5	1·041	5	27·68	25·0	1·209	23	−11·64
6·5	1·049	6	26·60	26·0	1·219	24	−17·14
8·0	1·058	7	25·52	27·0	1·229	25	−21·82
9·0	1·067	8	24·26	28·0	1·240	26	−27·04
10·0	1·076	9	22·82	29·0	1·250	27	−32·62
11·0	1·085	10	22·38	30·0	1·261	28	−39·28
12·0	1·094	11	19·76	31·0	1·272	29	−46·20
13·0	1·103	12	18·14	32·0	1·283	30	−54·40
14·5	1·112	13	16·34	33·0	1·294	31	−52·42
15·5	1·121	14	14·36	34·0	1·305	32	−39·28
17·0	1·131	15	12·20	35·0	1·316	33	−25·24
18·0	1·140	16	10·04	35·5	1·327	34	− 9·76
19·0	1·150	17	7·52	36·5	1·338	35	+ 2·84
20·0	1·159	18	4·64	37·5	1·349	36	+14·36

In commercial chloride of calcium there is a certain amount of water and impurities, so that approximately 30 per cent more calcium than tabulated is required for solutions having the specific gravities given.

TABLE B.

SIZES OF SINGLE-ACTING AMMONIA COMPRESSORS SUCH AS ARE MANUFACTURED BY THE YORK MFG. CO., YORK, PA.

	Tons Refrigerating Capacity per 24 Hours.	Compressor.		Engine.		Revolutions per Minute.	Fly-wheel.	
		Diameter.	Stroke.	Diameter	Stroke.		Diameter.	Weight.
A	2½	5½	6	6	8	126	2½	1,000
B	5	5½	6	8	8	126	2½	1,500
C	8	6½	8	9	10	110	3	2,000
D	12	7½	10	11½	10	100	4	3,000
E	20	9	12	13½	12	95	4½	4,000
F	30	11	15	16	15	77	5	5,000
G	45	12½	18	18	18	73	6	7,500
H	62½	14	21	20	21	70	7	10,000
I	90	16	24	24	24	67	8	12,000
J	125	18	28	26	28	64	9	15,000
K	175	20	32	28½	32	63	10	20,000
L	225	22½	36	32	36	57	12	32,000
M	300	25	42	36	42	53	14	40,000
N	500	30	48	44	48	53	16	60,000

	Diameter Crank Shaft.	Size Pipe Connections.			Space Required for Machine.			Weight Machine Complete.
		Suction and Discharge.	Steam.	Exhaust.	Length.	Width.	Height.	
A	4	1¼	1¼	1½	6′ 0″	2′ 4″	5′ 6″	3,600
B	4	1¼	1½	2	7′ 0″	2′ 4″	5′ 6″	4,800
C	5	1¼	2	2½	7′ 1″	3′ 2″	6′ 8″	8,800
D	5¾	1½	2	3	9′ 2″	6′ 9″	8′ 9″	14,200
E	6½	2	2½	3½	10′ 2″	7′ 2″	9′ 10″	18,000
F	7½	2½	3	3½	11′ 10″	7′ 11″	11′ 2″	34,000
G	9	3	3½	4½	13′ 8″	11′ 5″	12′ 7″	50,000
H	10	3½	4½	6	16′ 2″	12′ 2″	13′ 10″	65,000
I	12	4	6	7	17′ 10″	14′ 2″	15′ 3″	88,000
J	13	4	6	7	20′ 0″	14′ 6″	17′ 0″	105,000
K	14	5	7	8	22′ 11″	15′ 7″	19′ 5″	140,000
L	16	5	8	10	26′ 0″	17′ 3″	21′ 9″	185,000
M	17	6	10	12	30′ 0″	20′ 0″	25′ 3′	320,000
N	18	7	12	14	34′ 0″	20′ 1″	29′ 1″	425,000

TABLE C.

SIZES OF DOUBLE-ACTING AMMONIA COMPRESSORS SUCH AS ARE MANUFACTURED BY THE FRED. W WOLF CO., CHICAGO, ILL.

Compressor.						Engine.		Ammonia Condenser.			
Tons Refrigerating Capacity per 24 Hrs.	Diameter of Cylinder. Inches.	Stroke of Cylinder. Inches.	Size of Connections, Inches.	Horse-power Required.	Revolutions per Minute.	Diameter Stroke, Inches.	Floor Space Required for Compressor and Engine. Width. Length.	Number of Sections.	Number of Pipes.	Length of Pipes in Feet.	Space Required if Sections are placed 20-inch Centres. Length. Width. Height.
6	$5\frac{7}{8}$	$12\frac{5}{8}$	2	12	70	9×14	6′ 6″×11′ 9″	1	18	15	18× 2′ 0″× 9′ 3″
12	$8\frac{1}{16}$	$12\frac{5}{8}$	2	18	70	11×16	8′ 0″×12′ 6″	2	18	15	18× 3′ 9″× 9′ 3″
15	$8\frac{1}{4}$	15	2	23	70	11×16	8′ 0″×12′ 6″	2	18	$17\frac{1}{2}$	21× 3′ 9″× 9′ 3″
18	9	15	$2\frac{1}{2}$	27	70	10×30	9′ 0″×17′ 0″	2	18	20	24× 3′ 9″× 9′ 3″
20	$9\frac{1}{2}$	15	$2\frac{1}{2}$	30	70	10×30	9′ 6″×17′ 0″	2	20	20	24× 3′ 9″×10′ 6″
25	$9\frac{7}{8}$	$16\frac{5}{8}$	$2\frac{1}{2}$	38	70	12×30	9′ 6″×18′ 0″	2	24	20	24× 5′ 0″×12′ 0″
25	$10\frac{1}{4}$	17	$2\frac{1}{2}$	38	70	12×30	10′ 0″×18′ 6″	2	24	20	24× 5′ 0″×12′ 0″
33	11	$17\frac{5}{8}$	3	50	70	12×36	10′ 0″×20′ 6″	3	24	20	24× 6′ 8″×12′ 0″
40	11	$21\frac{1}{4}$	3	55	70	14×36	10′ 0″×20′ 6″	3	24	20	24× 6′ 8″×12′ 0″
40	11	$21\frac{1}{4}$	3	55	70	14×36	11′ 6″×20′ 6″	3	24	20	24× 6′ 8″×12′ 0″
50	$12\frac{13}{16}$	$21\frac{1}{4}$	3	63	66	16×36	11′ 6″×21′ 9″	4	24	20	24× 8′ 4″×12′ 0″
50	$12\frac{13}{16}$	$21\frac{1}{4}$	3	63	66	16×36	12′ 3″×21′ 9″	4	24	20	24× 8′ 4″×12′ 0″
65	$13\frac{3}{4}$	25	$3\frac{1}{2}$	81	63	16×42	12′ 6″×23′ 9″	5	24	20	24×10′ 0″×12′ 0″
75	14	30	$3\frac{1}{2}$	94	60	18×42	13′ 0″×25′ 0″	6	24	20	24×11′ 8″×12′ 0″
75	15	25	$3\frac{1}{2}$	94	60	18×42	12′ 3″×25′ 0″	6	24	20	24×11′ 8″×12′ 0″
85	15	30	4	106	57	20×42	13′ 9″×26′ 0″	7	24	20	24×13′ 4″×12′ 0″
90	$15\frac{1}{2}$	30	4	115	57	20×42	12′ 3″×26′ 0″	7	24	20	24×13′ 4″×12′ 0″
90	$15\frac{1}{2}$	30	4	115	57	20×42	14′ 0″×26′ 0″	7	24	20	24×13′ 4″×12′ 0″
100	$18\frac{1}{2}$	$27\frac{9}{16}$	5	125	50	22×42	13′ 6″×26′ 6″	8	24	20	24×15′ 0″×12′ 0″
100	$16\frac{1}{2}$	30	5	125	60	20×42	14′ 6″×26′ 6″	8	24	20	24×15′ 0″×12′ 0″
120	18	30	5	150	57	22×42	15′ 0″×26′ 6″	10	24	20	24×18′ 4″×12′ 0″
150	20	30	5	188	57	24×42	14′ 0″×27′ 0″	12	24	20	24×21′ 8″×12′ 0″
175	21	33	5	220	50	26×48	15′ 6″×29′ 0″	14	24	20	24×25′ 0″×12′ 0″
225	24	36	6	280	50	30×48	15′ 6″×30′ 9″	18	24	20	24×31′ 8″×12′ 0″
225	21	48	6	280	45	32×48	16′ 6″×31′ 0″	18	24	20	24×31′ 8″×12′ 0″
450	21 & 21	48 & 48	6	560	45	26×48×48	18′ 0″×44′ 3″	36	24	20	24×61′ 8″×12′ 0″
500	22 & 22	48 & 48	6	625	45	26×48×48	19′ 0″×44′ 3″	40	24	20	24×68′ 4″×12′ 0″
600	25 & 25	48 & 48	7	750	45	32×60×48	20′ 0″×44′ 6″	48	24	20	24×81′ 8″×12′ 0″

TABLE D.

RELATIVE HUMIDITY OF THE AIR, PER CENT, FAHRENHEIT TEMPERATURE, PRESSURE 30 INCHES.

Taken from the Weather Bulletin No. 235, U. S. Department of Agriculture, 1900.

Difference Between Wet- and Dry-bulb Thermometer.	Air Temperature.													
	–20°	–10°	0°	10°	20°	30°	32°	40°	50°	60°	70°	80°	90°	100°
0·1														
0·2	82	90	93	96	97									
0·3	73	84	90	93	95									
0·4	63	78	87	91	94									
0·5	54	73	83	89	92	94	95	96	96	97	98	98		
0·6	45	68	80	87	91									
0·7	37	62	76	84	89									
0·8	28	57	73	82	88									
0·9	19	51	70	80	86									
1·0	10	46	67	78	85	89	89	92	93	94	95	96	96	96
1·5	....	20	50	67	77	83	84	87	90	91	93	94		
2·0	....	....	33	56	70	78	79	83	87	89	90	91	92	93
2·5	....	....	17	45	62	73	74	79	83	86	88	89		
3·0	....	....	1	34	5	67	69	75	80	83	86	87	89	89
3·5	....	....	....	24	48	62	64	71	77	81	83	85		
4·0	....	....	....	13	40	56	59	68	74	78	81	83	85	86
4·5	....	....	....	....	33	51	54	64	71	75	79	81		
5·0	....	....	....	....	26	46	49	60	67	73	77	79	81	83
5·5	....	....	....	....	19	41	44	56	64	70	74	77		
6·0	....	....	....	....	12	36	39	52	61	68	72	75	78	80
6·5	....	....	....	....	5	31	35	48	58	65	70	74		
7·0	....	....	....	....	....	26	30	45	55	63	68	72	74	77
7·5	....	....	....	....	....	27	25	41	52	60	66	70		
8·0	....	....	....	....	....	16	20	37	49	58	64	68	71	73
8·5	....	....	....	....	....	11	16	33	46	55	61	66		
9·0	....	....	....	....	....	6	11	29	43	53	59	64	68	70
9·5	....	....	....	....	....	....	7	26	41	50	57	62		
10·0	....	....	....	....	....	....	....	22	38	48	55	61	65	68
10·5	....	....	....	....	....	....	....	18	35	46	53	59		
11·0	....	....	....	....	....	....	....	15	32	43	51	57	61	65
11·5	....	....	....	....	....	....	....	11	29	41	49	55		
12·0	....	....	....	....	....	....	....	7	27	39	48	54	58	62
12·5	....	....	....	....	....	....	....	....	24	37	46	52		
13·0	....	....	....	....	....	....	....	....	21	34	44	50	55	59
13·5	....	....	....	....	....	....	....	....	18	32	42	49		

TABLE D—*Continued.*

Difference Between Wet- and Dry-Bulb Thermometer.	Air Temperature.													
	–20°	–10°	0°	10°	20°	30°	32°	40°	50°	60°	70°	80°	90°	100°
14·0	....	....	....	....	....	....	....	....	16	30	40	47	52	56
14·5	....	....	....	....	....	....	....	....	13	28	38	45		
15·0	....	....	....	....	....	....	....	....	10	26	36	44	49	54
15·5	....	....	....	....	....	....	....	....	8	23	34	42		
16·0	....	....	....	....	....	....	....	....	....	21	33	41	47	51
16·5	....	....	....	....	....	....	....	....	....	19	31	39		
17·0	....	....	....	....	....	....	....	....	....	17	29	38	44	49
17·5	....	....	....	....	....	....	....	....	....	15	27	36		
18·0	....	....	....	....	....	....	....	....	....	13	25	35	41	46
18·5	....	....	....	....	....	....	....	....	....	11	24	33		
19·0	....	....	....	....	....	....	....	....	....	9	22	32	39	44
20·0	....	....	....	....	....	....	....	....	....	....	19	30	36	41
21·0	....	....	....	....	....	....	....	....	....	....	15	26	34	39
22·0	....	....	....	....	....	....	....	....	....	....	12	23	31	37
23·0	....	....	....	....	....	....	....	....	....	....	9	20	29	35
24·0	....	....	....	....	....	....	....	....	....	....	8	18	26	33
25·0	....	....	....	....	....	....	....	....	....	....	6	15	24	30

TABLE E.

SPECIFIC HEAT AND COMPOSITION OF VICTUALS.

From "Compend. of Mechanical Refrig."—Siebel.

	Water.	Solids.	Specific Heat Above Freezing, Degrees F.	Specific Heat Below Freezing, Degrees F.	Latent Heat of Freezing.
Lean beef	72·00	28·00	0·77	0·41	102
Fat "	51·00	49·00	0·60	0·34	72
Veal	63·00	37·00	0·70	0·39	90
Fat pork	39·00	61·00	0·51	0·30	55
Eggs	70·00	30·00	0·76	0·40	100
Potato	74·00	26·00	0·80	0·42	105
Cabbage	91·00	9·00	0·93	0·48	129
Carrots	83.00	17·00	0·87	0·45	118
Cream	59·25	30·75	0·68	0·38	84
Milk	87·50	12·50	0·90	0·47	124
Oysters	80·38	19·62	0·84	0·44	114
Whitefish	78·00	22·00	0·82	0·43	111
Eels	62·07	37·93	0·69	0·38	88
Lobster	76·62	23·38	0·81	0·42	108
Pigeon	72·40	27·60	0·78	0·41	
Chicken	73·70	26·30	0·80	0·42	

TABLE F.

TEMPERATURES ADAPTED FOR THE COLD STORAGE OF VARIOUS ARTICLES, FROM "REFRIGERATION, COLD STORAGE, AND ICE-MAKING," BY A. J. WALLIS THAYER.

Article.	Wallis Thayer.	Siebel.	Schmidt.	Getty.	Ice and Refrigeration.	Ice and Cold Storage.	Raue.	Madison Cooper.
Ale.	.....	33–42						
Apples.	32–36	33	32–36	32	32–36	33–36	.....	31
“ (summer)	.....	.....	.....	.....	.....	.....	38–42	
“ (winter)	.....	.....	.....	.....	.....	.....	32–35	
Apple- and peach-butter	.....	.....	.....	40	.....	.....	.....	42
Asparagus	34	34	34–35	34	34	35	34	33
Bananas	40–45	34–36	34	35	40–45	40–45	.....	36
Beans, dried.	32–40	.....	.....	32–34	.....	32–40	.....	40
Beef (fresh)	.....	.....	.....	37–39	.....	.....	.....	35
“ (dried)	.....	.....	.....	36–45				
Beer in casks or barrels	33–42	33–42	33–35	.....	32–42	33–40		
“ “ bottles	45	45	45	.....	45	45	.....	45
Berries, fresh (10 days)	36–40	.....	.....	35	.....	36–40	.....	40
Buckwheat flour	40	.....	.....	.....	.....	40	.....	42
Butter.	32–38	32–35	18–25	32–35	32–38	25–30	.....	14
“ to freeze.	.....	.....	.....	20				
Butterine.	35	.....	.....	.....	.....	35	.....	20
Cabbages.	34	.....	.....	33	.....	35	.....	32
Cantalopes	40	.....	.....	.....	.....	40	.....	40
Carrots.	34	.....	.....	34	.....	35	.....	33
Celery	32–34	33–35	34–35	34	32-34	35	.....	32
Cheese.	32–33	32–33	28–34	31–32	32–33	32–33	.....	35
Chestnuts.	33	.....	.....	33	.....	33	.....	40
Chocolate (to cool)	40							
Cider.	30–40	30–40	30–35	.....	30–40	35–40	.....	32
Cigars.	35	.....	.....	35	.....	35	.....	42
Clarets	.....	45–50						
Corn meal.	.....	.....	.....	.....	.....	.....	.....	42
“ (dried)	35	.....	.....	35	.....	35	.....	45
Cranberries.	34–36	.....	.....	.....	.....	34–36	.....	33
Cream.	35							
Cucumbers.	.....	.....	.....	.....	.....	.....	38–40	38
Currents	.....	.....	.....	.....	.....	.....	.....	32
Dates	55	.....	.....	.....	.....	55	.....	45
Eggs.	33–35	32–33	31–33	.....	32–35	33–35	.....	30
Ferns.	.....	.....	.....	.....	.....	.....	.....	28
Figs.	55	.....	.....	35				
Fish (fresh)	25–30	25–30	20	25	25–30	25–30	.....	28
“ (fresh-water, frozen)	.....	.....	.....	.....	.....	.....	.....	17
“ (salt-water frozen)	.....	.....	.....	.....	.....	.....	.....	14
“ (canned)	.....	.....	.....	35				
“ dried)	35	35	35–36	36	35	35	.....	40
“ to freeze.	.....	.....	.....	.....	.....	.....	.....	5
Flour, etc.	.....	40	36–40	40	40			
Fruits.	26–55							
“ (dried)	35–40	.....	.....	38	.....	35–40	.....	40
“ (canned).	35	.....	.....	35	.....	35	.....	30
Furs (undressed)	35	35	.....	35	35	.....	.....	35
“ (dressed)	25–32	25–32	28–35	25–32	25–32	35	.....	25
“ (long storage)	.....	.....	.....	.....	.....	.....	.....	10

TABLE F—*Continued.*

Article.	Wallis Thayer.	Siebel.	Schmidt.	Getty.	Ice and Refrigeration.	Ice and Cold Storage.	Raue.	Madison Cooper.
Grapes	36–38	32–40	34–36	32–40	36–38	36–80	38–40	34
Ginger ale	36	36	.....	.....	.....	35	.....	36
Hams	.....	.....	.....	30–35	.....	.....	.....	20
Hogs	.....	.....	.....	30–33				
Hops	33–40	33–36	32–40	35	33–40	35–40	.....	35
" (frozen)	.....	.....	.....	.....	.....	28		
Honey	45	36–40	45	37–40	45	45	.....	45
Lard	.....	.....	.....	34–35				
Lemons	36–40	36–45	33–36	35–45	36–40	36–40	.....	38
Liver	.....	.....	.....	30				
Maple-syrup sugar	.....	.....	.....	40–45	.....	.....	.....	45
Margarine	.....	35	18–25	.....	35			
Meat (brined)	.....	.....	.....	35–40				
" (canned)	35	.....	.....	35	.....	35	.....	30
" (fresh)	34	35	35–40	.....	34	35		
Melons (3 or 4 weeks)	.....	.....	.....	.....	.....	.....	.....	35
Milk	.....	.....	.....	32				
Mutton	.....	.....	.....	33–36				
" (frozen)	25–28							
Nuts in shells	35	35–38	35–40	35–38	35	35	.....	40
Oatmeat	40	.....	.....	.....	.....	40	.....	42
Oleomargarine	35	.....	.....	.....	.....	35	.....	20
Oil	35	.....	.....	35	.....	35	.....	45
Onions	34–40	32–33	36	32	34–40	35–40	34–40	32
Oranges	45–50	36	34–36	35–36	45–50	45–50	.....	34
Oysters	33–35	.....	.....	.....	.....	33–35	.....	33
" (in tubs)	25	.....	.....	35	.....	25	.....	35
" (in shells)	33	.....	.....	40	.....	30	.....	43
Ox-tails	.....	.....	.....	32				
Parsnips	34	.....	.....	34	.....	35	.....	32
Peaches	45–55	35–45	34–36	35–45	45–55	45–55	36–38	36
Pears (dried)	40	.....	.....	40	.....	40	.....	45
Plums	.....	.....	.....	.....	.....	.....	38–40	32
Porter	.....	33–42						
Pork	34							
Potatoes	36–40	34–36	.....	35	36–40	36–40	36–40	34
Poultry (frozen)	28–30	28–30	20–28	29	28–30	28–30	.....	28
" (to freeze)	18–22	.....	.....	5–10	.....	18–22		
" (long storage)	.....	.....	.....	.....	.....	.....	.....	10
Sardines (canned)	35	.....	.....	.....	.....	35	.....	40
Sauerkrout	35–38	.....	.....	36–38	.....	35–38	.....	38
Sausage (casings)	.....	.....	.....	30–35	.....	.....	.....	20
Sugar, etc.	40–45	.....	.....	.....	.....	40–45	.....	45
Syrup	35	.....	.....	35	.....	35	.....	45
Tenderloin butts, ribs, etc.	.....	.....	.....	30–35				
Tomatoes	.....	34–35	38–42	35	.....	35	38–42	42
Tobacco	35	.....	.....	35	.....	35	.....	42
Veal	.....	.....	.....	32–36				
Vegetables	34–40							
Watermelons	34	.....	.....	.....	.....	35	.....	40
Wheat flour	40	.....	.....	.....	.....	40	.....	42
Wines	40–45	40–45	40–45	.....	40–45	45–50	.....	42
Woollens	25–32	25–32	28–35	25–32	25–32	25–30		

TABLE G.

PROPERTIES OF THE SATURATED VAPOUR OF AMMONIA.

Temperature.		Pressure, Absolute, p.		Heat of Vapourization, Thermal Units. Eq. (360). h_c.	External Heat, Thermal Units. $\frac{p(v-v_1)}{J}$.	Internal Heat, Thermal Units. $p=h_c-\frac{pv}{J}$.	Volume of Vapour per Pound, Cubic Foot. Eq. (362). v.	Volume of Liquid per Pound, Cubic Foot. v_1.	Weight of a Cubic Foot of Vapour, Pounds. $\frac{1}{v}$.
Degree F. T.	Absolute. τ.	Pounds per Square Foot. Eq. (350).	Pounds per Square Inch.						
−40	420·66	1540·7	10·69	579·67	48·23	531·44	24·372	·0234	·0410
−35	425·66	1773·6	12·31	576·69	48·48	528·21	21·319	·0236	·0468
−30	430·66	2035·8	14·13	573·69	48·77	524·92	18·697	·0237	·0535
−25	435·66	2329·5	16·17	570·68	49·06	521·62	16·445	·0238	·0608
−20	440·66	2657·5	18·45	567·67	49·38	518·29	14·507	·0240	·0689
−15	445·66	3022·5	20·99	564·64	49·67	514·97	12·834	·0242	·0779
−10	450·66	3428·0	23·80	561·61	49·99	511·62	11·384	·0243	·0878
− 5	455·66	3877·2	26·93	558·56	50·31	508·25	10·125	·0244	·0988
0	460·66	4373·5	30·37	555·50	50·68	504·82	9·027	·0246	·1108
+ 5	465·66	4920·5	34·17	552·43	50·84	501·59	8·069	·0247	·1239
+10	470·66	5522·2	38·34	549·35	51·13	498·22	7·229	·0249	·1383
+15	475·66	6182·4	42·93	546·26	51·33	494·93	6·492	·0250	·1544
+20	480·66	6905·3	47·95	543·15	51·61	491·54	5·842	·0252	·1712
+25	485·66	7695·2	53·43	540·03	51·80	488·23	5·269	·0253	·1898
+30	490·66	8556·6	59·41	536·92	52·01	484·91	4·763	·0254	·2100

Table H.

PROPERTIES OF AIR.

Of the Weights of Air, Vapour of Water, and Saturated Mixtures of Air and Vapour of Different Temperatures, under the Ordinary Atmospheric Pressure of 29.921 Inches of Mercury.

Temperature, Deg. Fahr.	Volume of Dry Air at Different Temperatures, the Volume at 32° being 1000.	Weight of a Cubic Foot of Dry Air at Different Temperatures in Pounds.	Elastic Force of Vapour in Inches of Mercury (Regnault).	Mixtures of Air Saturated with Vapour.			
				Elastic Force of the Air in the Mixture of Air and Vapour in Inches of Mercury.	Weight of a Cubic Foot of the Mixture.		
					Weight of the Air in Pounds.	Weight of the Vapour in Pounds.	Total Weight of Mixture in Pounds.
1	2	3	4	5	6	7	8
0	·935	·0864	0·044	29·877	·0863	·000079	·086379
12	·960	·0842	·074	29·849	·0840	·000130	·084130
22	·980	·0824	·118	29·803	·0821	·000202	·082302
32	1·000	·0807	·181	29·740	·0802	·000304	·080504
42	1·020	·0791	·267	29·654	·0784	·000440	·078840
52	1·041	·0776	·388	29·533	·0766	·000627	·077227
60	1·057	·0764	·522	29·399	·0751	·000830	·075252
62	1·061	·0761	·556	29·365	·0747	·000881	·075581
70	1·078	·0750	·754	29·182	·0731	·001153	·073509
72	1·082	·0747	·785	29·136	·0727	·001221	·073921
82	1·102	·0733	1·092	28·829	·0706	·001667	·072267
92	1·122	·0720	1·501	28·420	·0684	·002250	·070717
100	1·139	·0710	1·929	27·992	·0664	·002848	·069261
102	1·143	·0707	2·036	27·885	·0659	·002997	·068897
112	1·163	·0694	2·731	27·190	·0631	·003946	·067042
122	1·184	·0682	3·621	26·300	·0599	·005142	·065046
132	1·204	·0671	4·752	25·169	·0564	·006639	·063039
142	1·224	·0660	6·165	23·756	·0524	·008473	·060873
152	1·245	·0649	7·930	21·991	·0477	·010716	·058416
162	1·265	·0638	10·099	19·822	·0423	·013415	·055715
172	1·285	·0628	12·758	17·163	·0360	·016682	·052682
182	1·306	·0618	15·960	13·961	·0288	·020536	·049336
192	1·326	·0609	19·828	10·093	·0205	·025142	·045642
202	1·347	·0600	24·450	5·471	·0109	·030545	·041445
212	1·367	·0591	29·921	0·000	·0000	·036820	·036820

Table H—*Continued.*

PROPERTIES OF AIR.

Temperature, Deg. Fahr.	Mixture of Air Saturated with Vapor.		Cubic Feet of Vapour from One Pound of Water at Pressure as in Column 4.	B.T.U. Absorbed by One Cubic Foot Dry Air per Degree Fahr.	B.T.U. Absorbed by One Cubic Foot Saturated Air per Degree Fahr.	Cubic Feet Dry Air Warmed One Degree per B.T.U.	Cubic Feet Saturated Air Warmed One Degree per B.T.U.
	Ratio of Water to Dry Air.	Ratio of Dry Air to Water Vapour.					
1	9	10	11	12	13	14	15
0	·00092	1092·4	....	·02056	·02054	48·5	48·7
12	·00115	646·1	....	·02004	·02006	50·1	50·0
22	·00245	406·4	....	·01961	·01963	51·1	51·0
32	·00379	263·81	3289	·01921	·01924	52·0	51·8
42	·00561	178·18	2252	·01882	·01884	53·2	52·8
52	·00819	122·17	1595	·01847	·01848	54·0	53·8
60	·01251	92·27	1227	·01818	·01822	55·0	54·9
62	·01179	84·79	1135	·01811	·01812	56·2	55·7
70	·01780	64·59	882	·01777	·01794	57·3	56·5
72	·01680	59·54	819	·01777	·01790	58·5	56·8
82	·02361	42·35	600	·01744	·01770	57·2	56·5
92	·03289	30·40	444	·01710	·01751	58·5	57·1
100	·04495	23·66	356	·01690	·01735	59·1	57·8
102	·04547	21·98	334	·01682	·01731	59·5	57·8
112	·06253	15·99	253	·01651	·01711	60·6	58·5
122	·08584	11·65	194	·01623	·01691	61·7	59·1
132	·11771	8·49	151	·01596	·01670	62·5	59·9
142	·16170	6·18	118	·01571	·01652	63·7	60·6
152	·22465	4·45	93·3	·01544	·01654	65·0	60·5
162	·31713	3·15	74·5	·01518	·01656	62·2	60·4
172	·46338	2·16	59·2	·01494	·01658	67·1	60·3
182	·71300	1·402	48·6	·01471	·01687	68·0	59·5
192	1·22643	·815	39·8	·01449	......	68·9	
202	2·80230	·357	32·7	·01466	......	68·5	
212	Infinite	·000	27·1	·01406	......	71·4	

TABLE I.

WEIGHT OF A CUBIC FOOT OF AQUEOUS VAPOUR AT DIFFERENT TEMPERATURES AND PERCENTAGES OF SATURATION.

Temp., ° F.	Percentage of Saturation.									
	10	20	30	40	50	60	70	80	90	100
	Grains	Grains	Grains	Grains	Grains	Grains	Grains	Grains	Grains	Grains
−20	0·017	0·033	0·050	0·066	0·083	0·100	0·116	0·133	0·149	0·166
−19	0·017	0·035	0·052	0 070	0·087	0·104	0·122	0·139	0·157	0·174
−18	0·018	0·037	0·055	0·074	0·092	0·110	0·129	0·147	0·166	0·184
−17	0·020	0·039	0·059	0·078	0·098	0·118	0·137	0·157	0·176	0·196
−16	0·021	0·041	0·062	0·083	0·104	0·124	0·145	0·166	0·186	0·207
−15	0·022	0·044	0·065	0·087	0·109	0·131	0·153	0·174	0·196	0·218
−14	0·023	0·046	0·069	0·092	0·116	0·139	0·162	0·185	0·208	0·231
−13	0·024	0·049	0·093	0·097	0·122	0·146	0·170	0·194	0·219	0·243
−12	0·026	0·051	0·077	0·103	0·128	0·154	0·180	0·206	0·231	0·257
−11	0·027	0·054	0·081	0·108	0·135	0·162	0·189	0·216	0·243	0·270
−10	0·028	0·057	0·086	0·114	0·142	0·171	0·200	0·228	0·256	0·285
− 9	0·030	0·060	0·090	0·120	0·150	0·180	0·210	0·240	0·270	0·300
− 8	0·032	0·063	0·095	0·126	0·158	0·190	0·221	0·253	0·284	0·316
− 7	0·033	0·066	0·100	0·133	0·166	0·199	0·232	0·266	0·299	0·332
− 6	0·035	0·070	0·105	0·140	0·175	0·210	0·245	0·280	0·315	0·350
− 5	0·037	0·074	0·111	0·148	0·185	0·222	0·259	0·296	0·333	0·370
− 4	0·039	0·078	0·117	0·156	0·194	0·233	0·272	0·311	0·350	0·389
− 3	0·041	0·082	0·123	0·164	0·206	0·247	0·288	0·329	0·370	0·411
− 2	0·043	0·087	0·130	0·174	0·217	0·260	0·304	0·347	0·391	0·434
− 1	0·046	0·091	0·137	0·183	0·228	0·274	0·320	0·366	0·411	0·457
0	0·048	0·096	0·144	0·192	0·240	0·289	0·337	0·385	0·433	0·481
+ 1	0·050	0·101	0·152	0·202	0·252	0·303	0·354	0·404	0·454	0·505
2	0·053	0·106	0·159	0·212	0·264	0·317	0·370	0·423	0·476	0·529
3	0·055	0·111	0·166	0·222	0·277	0·332	0·388	0·443	0·499	0·554
4	0·058	0·116	0·175	0·233	0·291	0·349	0·407	0·466	0·524	0·582
5	0·061	0·122	0·183	0·244	0·305	0·366	0·427	0·488	0·549	0·610
6	0·064	0·128	0·192	0·256	0·320	0·383	0·447	0·511	0·575	0·639
7	0·067	0·134	0·201	0·268	0·336	0·403	0·470	0·537	0·604	0·671
8	0·070	0·141	0·211	0·282	0·352	0·422	0·493	0·563	0·634	0·704
9	0·074	0·148	0·222	0·296	0·370	0·443	0·517	0·591	0·665	0·739
10	0·078	0·155	0·233	0·310	0·388	0·466	0·543	0·621	0·698	0·776
11	0·082	0·163	0·245	0·326	0·408	0·490	0·571	0·653	0·734	0·816
12	0·086	0·171	0·257	0·342	0·428	0·514	0·599	0·685	0·770	0·856
13	0·090	0·180	0·269	0·359	0·449	0·539	0·629	0·718	0·808	0·898
14	0·094	0·188	0·282	0·376	0·470	0·565	0·659	0·753	0·847	0·941
15	0·099	0·197	0·296	0·394	0·493	0·592	0·690	0·789	0·887	0·986
16	0·103	0·206	0·310	0·413	0·516	0·619	0·722	0·826	0·929	1·132
17	0·108	0·216	0·324	0·432	0·540	0·648	0·756	0·864	0·972	1·080
18	0·113	0·226	0·338	0·451	0·564	0·677	0·790	0·902	1·015	1·128
19	0·118	0·236	0·354	0·472	0·590	0·709	0·827	0·945	1·063	1·181

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